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PART IV
ASTIA DOCUMENT No. AD 130939

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STUDY OF MINIATURE ENGINE GENERATOR SETS

Part IV Investigation of Altitude and Low Temperature Performance; Starting, Cooling, Carburetion, Controls Systems; and Noise Reduction

OWEN E. BUXTON JR.
RICHMOND A. GOODEN
ARTHUR W. LEISSA
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THE OHIO STATE UNIVERSITY RESEARCH FOUNDATION

OCTOBER 1956

WRIGHT AIR DEVELOPMENT CENTER

NOTICES

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EQUIPMENT LABORATORY
CONTRACT NO. AF 18(600)-192
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WRIGHT AIR DEVELOPMENT CENTER
AIR RESEARCH AND DEVELOPMENT COMMAND
UNITED STATES AIR FORCE
WRIGHT-PATTERSON AIR FORCE BASE, OHIO

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FOREWORD

This report was prepared in the Mechanical Engineering Department of The Ohio State University under Contract No. AF 18(600)-192, "Study of Miniature Engine-Generator Sets" with The Ohio State University Research Foundation. Work was conducted under Project No. 6-(1-6058), "Electrical Generation Equipment," Task No. 60266, "Development of Minature Engine-Generator Sets," (formerly RDO No. 656-2112), and was directed by the Equipment Laboratory, with Dr. Edwin Naumann and Lt. R. G. Leiby serving as project engineers.

This report covers the period from January 1954, through September 1956, and is a continuation of the work reported in WADC TR 53-180 (May 1953), WADC TR 53-180 Part II (December, 1954) and WADC TR 53-180 Part III (March 1956). This report deals with additional work performed on the engines and does not include any additional work concerning the generators. The study will be completed with Part V - "Summary Report: Feasibility". October 1956.

ABSTRACT

This report deals with various performance parameters associated with miniature engines. Performances under low temperature and low pressure ambients are discussed. Design considerations and suitabilities dealing with starting, cooling, carburetion, and control systems are presented. Results of exploratory tests on noise control, as well as recent engine experience on two new test engines, concludes this period report on the project.

PUBLICATION REVIEW

The publication of this report does not constitute approval by the Air Force of the findings or the conclusions contained therein. It is published only for the exchange and stimulation of ideas.

FOR THE COMMANDER:

J. E. Rowland

J. E. ROWLAND
Chief, Systems & Power
Conversion Section
Electrical Branch
Equipment Laboratory

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INTRODUCTION

Miniaturization and weight reduction of mechanical and electronic equipment have become increasingly important over the past decade. This is particularly true of equipment with intended military or airborne applications where size and weight are of major logistic and economic importance.

The purpose of this project has been to determine the feasibility and general background information associated with miniature engine-generator sets. Units of the size of 35 to 400 watts were under consideration, it being felt that such a power source would offer tremendous possibilities if its performance and reliability could approach that of the larger engine-generator sets.

This report, which is the fourth of the overall study, summarizes part of the investigation taking place on the engine alone from the period January 1955 to October 1956. The details of the experiences obtained during the evaluation of various component systems is described as well as information obtained from continued testing of miniature engines. The systems discussed include starting, cooling, carburetion, and controls. Other information is included on altitude and low temperature performance as well as some exploratory studies into the problem of noise reduction.

WADC TR 53-180 (May 1953) describes the effort during the first year of the project. It describes the special test apparatus and instrumentation which had to be developed to conduct an experimental program with this miniature equipment. A discussion of some of the typical malfunctions and part failures which were experienced, as well as specification and performance curves for commercial model engines, are included. A short study is also presented to provide an approximate comparison between batteries and potential miniature engine-generator sets on a weight and volume basis.

WADC TR 53-180 Part II (December, 1954) presents the results of the experimental program conducted from May 1953 through December 1954. It deals with the performance characteristics of miniature engines, and the fuel and lubricant requirements peculiar to them. Most of the experimental data presented in this second report were obtained from special miniature engines designed, built and developed by the personnel of this project specifically as test vehicles for this study. A discussion of some optimum design features and limitations of these engines in reliability is included.

Manuscript released by the author 11 September 1956 for publication as a WADC Technical Report.

WADC TR 53-180 Part III (March 1956), the third report of this project, presents design and performance information concerning miniature dc generators. These data were obtained experimentally by this project, and include results for some miniature commercial generators as well as for some special types designed and built for experimental purposes. A design procedure is included to aid in the design of new miniature dc generators for specific applications.

WADC TR 53-180 Part V (October 1956), the final report of this project, is being prepared simultaneously with this report. The final report is intended to summarize the findings of this project during the four years of investigation. It deals with the potential capabilities of miniature engines and generators and with the reliability which might be expected. Conclusions reached regarding optimumly designed features of such units are discussed.

1. ALTITUDE AND LOW TEMPERATURE PERFORMANCE

Altitude and low temperature performance tests were conducted in order to determine the feasibility of operation for a miniature engine-generator set at extreme environmental conditions. Information concerning these effects was obtained for both two-cycle and four-cycle engines, and the performance results for each variable taken independently is given below. No attempt was made to study the combined effect of temperature and altitude; however, it is felt that such performance should be predictable from the independent studies.

All altitude and cold performance tests were conducted in an environmental testing chamber in which temperature, pressure, and humidity could be controlled. A dc electric motor rated at 1/3 hp was used to turn the engine over for starting purposes. Exhaust gases from the engine were directly evacuated without contaminating the chamber; however, the slight amount of suction required at the exhaust manifold to accomplish this was not sufficiently great to affect engine performance. All tests were run using either methyl alcohol or aviation gasoline as the basic fuel. The fuel was kept inside the chamber so as to be at the same temperature as the engine environment and to avoid the development of a positive fuel head in altitude tests as the chamber pressure was decreased. Spark ignition was used throughout.

1.1 ALTITUDE PERFORMANCE

Three engines were used to evaluate altitude effects, one four-cycle engine and two two-cycle engines. The four-cycle engine used was the L-head model described in Table 2 from "Study of Miniature Engine-Generator Sets, Part II" (Ref. 35). The two-cycle engines used were the Ruckstell-Hayward Pockette and the Mark III engines, both described in this report in the section entitled "Recent Engine Experience." Tests were primarily conducted to determine the maximum altitude at which these engines could be operated without supercharging.

The optimum needle valve setting was first determined by adjustment with the engine running until the maximum speed possible was obtained. Then the engine was started at atmospheric pressure with the door of the environmental chamber closed; thereafter, the chamber was gradually evacuated until the engine stopped. Then the test was repeated for needle valve settings which gave various degrees of richness or leanness at atmospheric pressure.

In every case, the needle valve setting required for maximum altitude was found to be one giving a slightly lean mixture at atmospheric conditions. This is because of the fact that the amount of air passing through the intake venturi is decreased at higher altitudes whereas the rate of fuel injection changes relatively little, resulting in a richening of the mixture as altitude is increased. This corroborates the experience found in operating reciprocating aircraft engines.

The L-head, four-cycle engine was operated with both 80-octane aviation gasoline and methyl alcohol as fuels. Using aviation gasoline, the engine operated to a point corresponding to an altitude of 22,000 feet, under the best conditions of needle valve setting. With methyl alcohol, the engine could reach only the equivalent of 17,000 feet altitude.

The Mark III, two-cycle engine was able to operate to a point equivalent to 17,000 feet using a mixture of 90 per cent aviation gasoline and 10 per cent SAE No. 20 lubricating oil. Using a mixture consisting of 90 per cent methyl alcohol and 10 per cent castor oil, it was able to operate up to 19,000 feet. The Ruckstell-Hayward Pockette unit was operated only on a gasoline-oil mixture and reached 15,000 feet before stopping.

These results indicate that four-cycle engines will give better altitude performance with gasoline than with methyl alcohol, whereas the contrary is true for two-cycle engines. Under normal operating conditions, two-cycle engines can be made to run faster and produce more power with alcohol than with gasoline, largely because of the high volatility of alcohol which allows better mixtures to reach the combustion chamber at the high speeds involved. This effect is less predominant for four-cycle engines, again because the speed is less. Therefore, under conditions of altitude, the low pressure air into which the fuel is injected allows better vaporization and hence the advantages of alcohol are diminished reltaive to gasoline.

Four-cycle engines, being more stable in operation at normal conditions than two-cycle engines, should lead one to believe that the former would also operate better at altitude conditions. The data described above verifies this. Since the lack of air is the primary reason for the engine to cease operating at high altitudes, the positive displacement scavenging of the four-cycle engine becomes exceedingly important.

Tests were also conducted with the L-head, four-cycle engine and the Mark III, two-cycle engine to determine the maximum altitude at which they could be started. Using 80 octane aviation gasoline as fuel, it was found that the L-head engine was able to start at altitudes as high as 9,000 feet; using methyl alcohol, it was able to start up to 8,000 feet. The Mark III showed better starting characteristics, being able to start at 11,000 feet using a 90-10 gasoline-oil mixture and at 14,000 feet using a 90-10 methanol-castor oil mixture.

The behavior of a miniature engine as the environmental operating altitude is increased is shown in Figures 1 and 2. Figure 1 describes the engine operation with a needle valve setting which gives optimum performance at ground-level conditions. The four-cycle engine operates at its maximum speed of 11,000 rpm for a brief interval and then the speed falls off rapidly to the vicinity of 8,000 rpm. Then stable operation is maintained at a virtually constant speed for most of the duration of the test. Finally, the engine speed again falls off rapidly as the maximum operating altitude is reached.

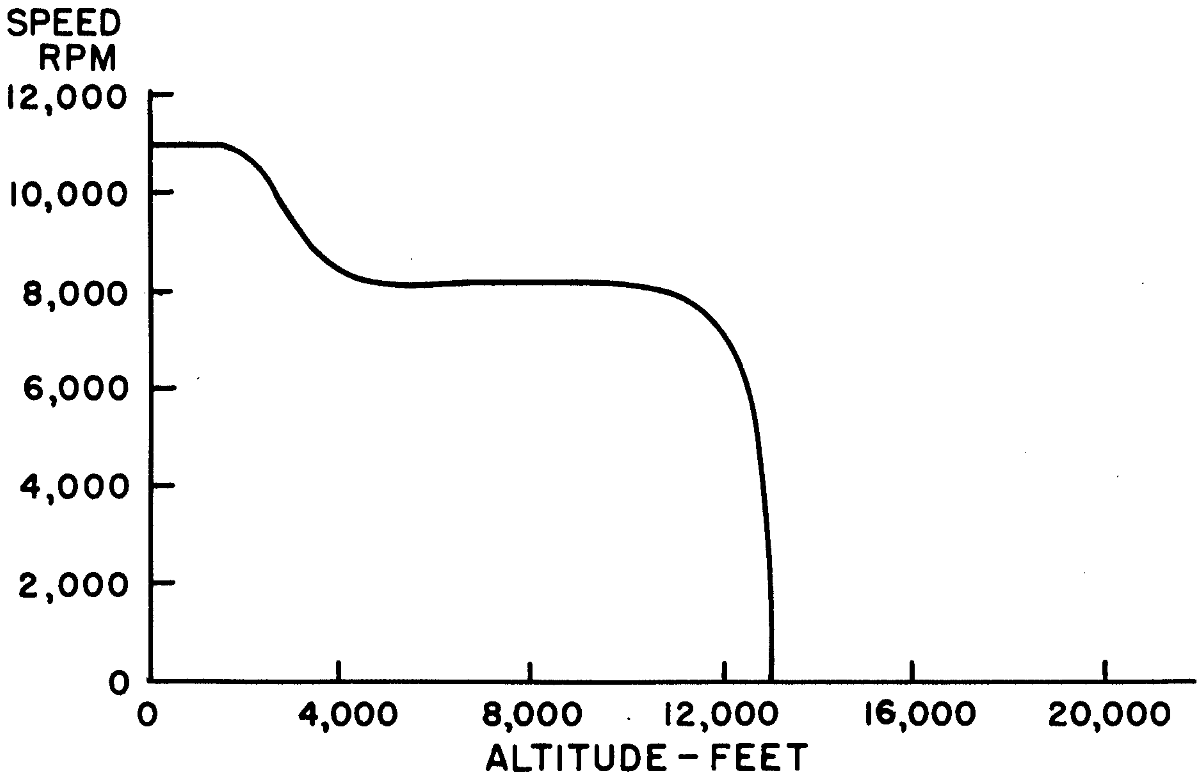


Fig. 1. Behavior of a Four-Cycle Engine at Various Altitudes with Optimum Air-Fuel Mixture for Ground-Level Operation

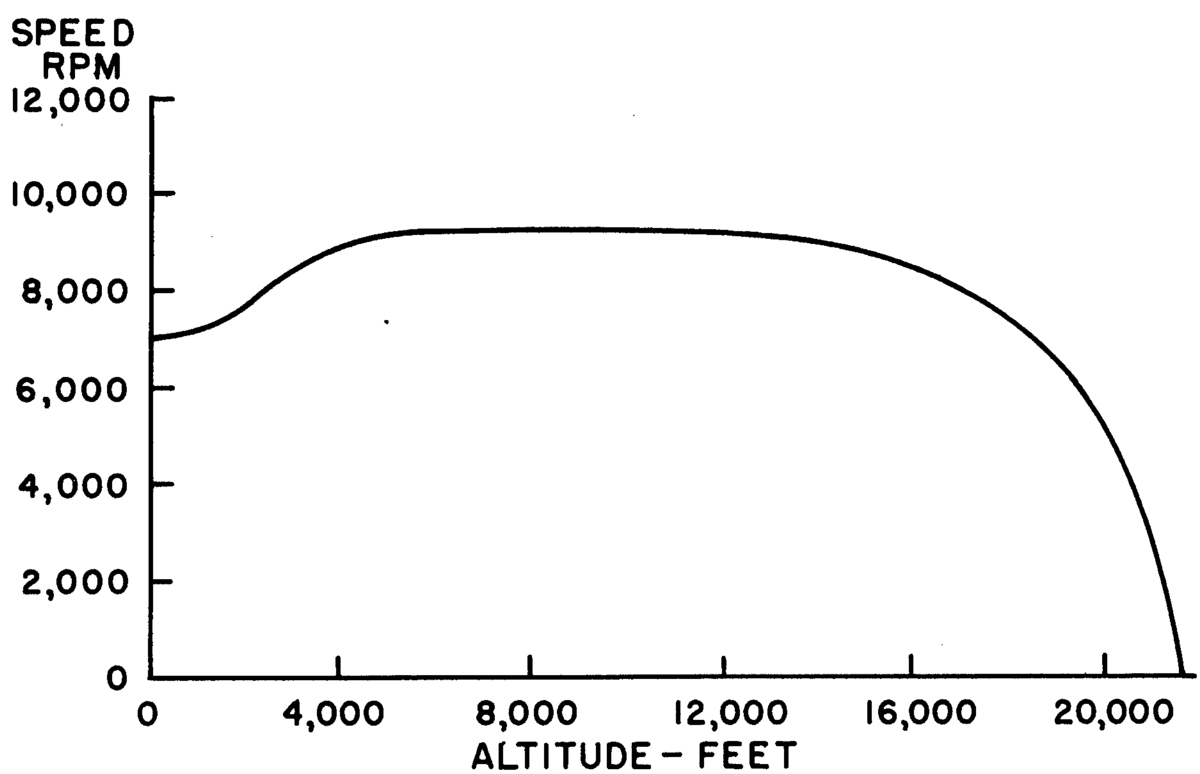


Fig. 2. Behavior of a Four-Cycle Engine at Various Altitudes with Lean Air-Fuel Mixture for Ground-Level Operation

Figure 2 shows the engine performance with a needle valve setting which is slightly lean for ground-level operation. The engine operates at only 7,000 rpm at ground-level conditions, but as the air pressure is decreased, the air-fuel mixture richens and causes an increase in speed to 9,000 rpm. After maintaining stable operation for a long period at this speed, the engine finally slows down and stops as the maximum operating altitude is reached; however, the engine does not die out as quickly as in the case described in Figure 1.

Figures 1 and 2 pertain to the operation of the L-head, four-cycle engine. Very similar curves may be drawn for the two-cycle engines, for the behavior was generally the same. For example, the Mark III engine would begin operating at a speed of 17,000 rpm at optimum ground-level mixture setting and then soon fall off to about 12,000 rpm for stable operation.

No attempt was made to place any external loading on the engines during the altitude testing. However, the engines were operating at a high frictional load due to the attached dc electric motor and spark ignition system as well as the inherent frictional load of the engine itself. It is estimated that, due to these conditions, the engines were operating at approximately 1/3 full load during the tests.

1.2 LOW TEMPERATURE PERFORMANCE

Performances of miniature engines in cold environments have been analyzed to determine the effect of reduced temperature upon engine operating characteristics and the ease of starting. Tests conducted with the Ruckstell-Hayward Beacon unit over a temperature range of 85°F to -9°F indicated that the change in power output was negligible. Other tests have even shown a slight increase in power output at cold temperatures (Ref. 5). The increased air density at cold temperature makes engine breathing easier. Also, operating difficulties are minimized owing to the fact the single-cylinder miniature engine reduces fuel distribution problems. Therefore, cold climate operation of miniature engines seems very feasible.

The starting of miniature engines in a cold environment presents many problems, however. Basically, the most important single factor in limiting the ease of starting is the inherent problem of volatilizing and distributing the fuel. The problem becomes more difficult as the environmental temperature is lowered.

Two variables must be considered in attempting quantitatively to describe ease of starting. The first variable is the length of time for which the engine is cranked over before it starts; the second is the speed required to crank the engine over to insure starting. The air-fuel mixture is heated during each compression stroke of the engine. The heat thus generated while the engine is being cranked over is then partly lost through the exhaust of the unburned mixture and is also partly absorbed by the cylinder wall. After many such compression strokes, the cylinder wall gradually becomes heated, with a temperature gradient extending throughout

the engine. As a part of this general engine temperature rise, the intake manifold also becomes warmer. Thus, owing to this effect of heating the manifold and the cylinder wall, combustion is encouraged until finally a sufficient amount of the mixture burns to allow the engine to drive itself.

However, if the engine were cranked over very slowly, the cylinder wall would absorb heat at a lesser rate and, in turn, the intake manifold would be colder, and ignition might never take place. This makes the speed of cranking the engine over of prime importance. Experience with miniature engines has shown that this second variable of cranking speed is by far the most important. It has been found that a 20 per cent increase in cranking speed may decrease the time required to start the engine to one-tenth of its former value. For example, it may take 20 seconds of cranking at 2,000 rpm to start an engine at a given temperature, but an increase of speed to 2,400 rpm may require only two seconds of cranking time. The importance of engine cranking speed in determining ease of starting has also been discovered in the use of reciprocating aircraft engines (Ref. 18).

Having determined engine cranking speed to be the single most important variable affecting ease of starting, tests were conducted on the L-head, four-cycle engine and the Mark III, two-cycle engine to study this effect. Necessary starting speeds were determined for a range of temperatures from 70°F to -10°F in some cases. The engine was cranked over with its ignition switch on by means of a dc electric motor. The rotary acceleration was made to be high at the beginning of the cranking and decreased near the vicinity of the starting speed. This allowed from one to three seconds for the engine to reach its starting speed, thus avoiding excess heating of the cylinder walls due to a long cranking period and yet allowing the speed to be read accurately on an electronic tachometer. The instant at which the engine started could be determined by any of three indications: (1) a sudden decrease in armature current required by the electric motor at the instant that the engine reversed the torque or (2) a sudden spurt in speed indicated by the tachometer dial or (3) the sound of the engine firing. The temperature for each starting test was measured by means of a thermocouple inserted under the spark plug so as to be close to the cylinder. Sufficient delay was allowed between each test to allow the engine temperature to stabilize near the chamber temperature.

It was necessary to choke the engine to insure starting, especially at low temperatures. To study objectively the effect of choking on starting, the air intake and needle valve assembly was adapted with a removable choking orifice as is shown in Figure 3.

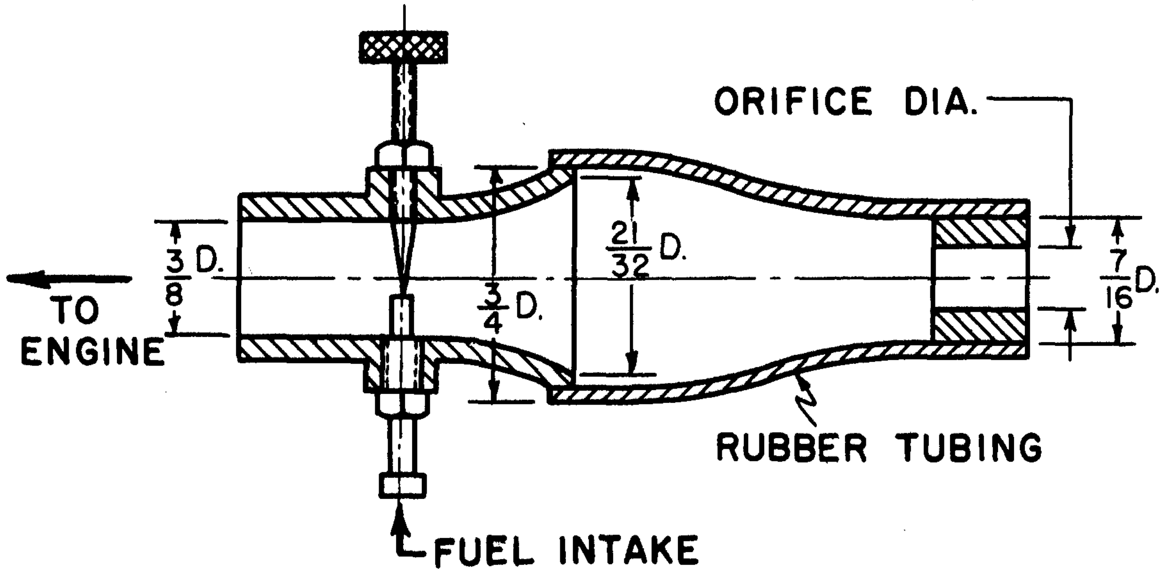


Figure 3. Choking Arrangement

A series of starting tests were run with the L-head, four-cycle engine using 80 octane aviation gasoline with the following diameters of choke orifices, in inches: 5/16, 1/4, 3/16, 1/8, 1/16. The results obtained with these choke settings are shown in the curves of Figure 4.

These curves show that the required starting speed increases as the engine temperature is decreased. Also, the slope of each curve increases as the temperature is decreased. The curves do not appear to be asymptotic to any given temperature. If they were, this would indicate a limiting temperature below which the engine would not start, regardless of speed. Nevertheless, the lowest recorded temperature plotted for each choice setting was the practical limit determined beyond which it was impossible to start the engine.

The curves indicate that the 1/8-inch diameter orifice opening is the best size to be used for the geometry of this particular engine, regardless of temperature conditions, since the required starting speed is the least with this size for every temperature. No curve is shown for the 1/16-inch diameter orifice because the engine would not start at all with this size.

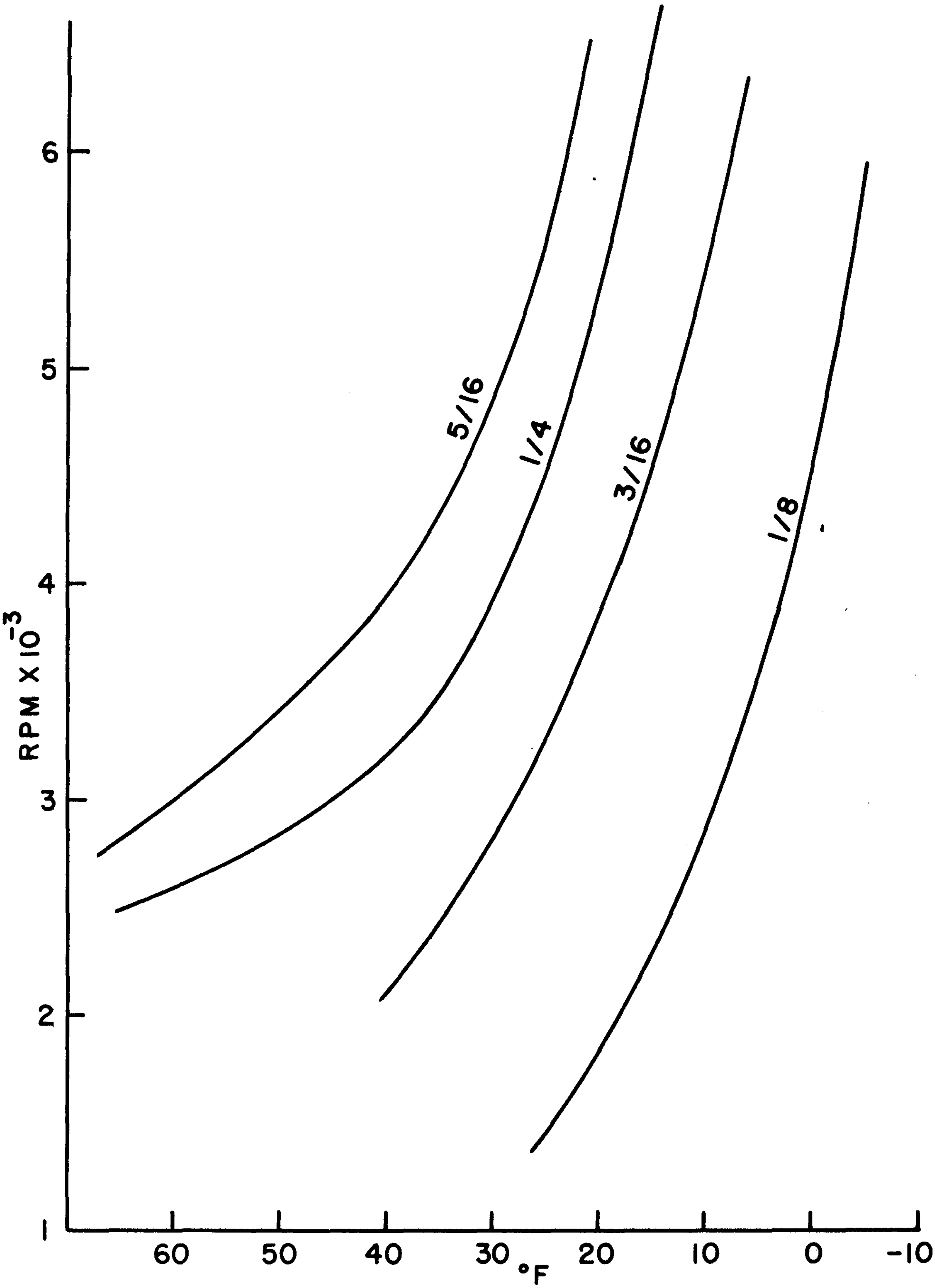


Figure 4. Minimum Starting Speed vs. Engine Temperature for Various Size Choke Orifice Diameters, for Four-Cycle L-Head Engine, Using Aviation Gasoline Fuel

The data from which these curves were plotted had very little scatter. Several values of starting speed were obtained for each point plotted, and the average of these was used. The deviation from the average was never greater than 200 rpm, indicating that the criterion of starting speed gives consistent results.

Tests were also run with the L-head, four-cycle engine using methyl alcohol as a fuel. Figure 5 represents the best data obtained with methyl alcohol as a fuel compared with that of aviation gasoline. The curves plotted for the 1/8-inch and 1/16-inch diameters of choke orifice for methyl alcohol indicate that these two sizes are virtually equally good. The fact that the engine would start well with the 1/16-inch diameter choke using alcohol and not when gasoline was used is due to the fact that alcohol requires a lower air-fuel ratio. Comparison of the curves for methyl alcohol with that shown for aviation gasoline indicates easier starting with the latter. One reason suggested for this phenomena is that even though methyl alcohol is considerably more volatile than gasoline, its latent heat of vaporization is more than three times as great as that for gasoline. This suggests that a high evaporative cooling effect in the manifold and cylinder is the cause of starting difficulty. This phenomena has been encountered in starting tests with other types of internal combustion engines using alcohol as a fuel (Ref. 24). In cases where engines are to be started and operated in cold climates using alcohol as a fuel, some means must be provided to insure starting. One method would be to provide a dual fuel system employing gasoline for starting and alcohol for later operation. Another method is to externally heat the intake manifold before starting. To insure good operation after starting at cold temperatures, a method of recycling a portion of the heated exhaust gas into the carburetor has been found to work well, although a slight loss in power must be consequently expected (Ref. 24).

Starting tests were also attempted with the Mark III, two-cycle engine using both aviation gasoline and methyl alcohol as fuels. However, it was impossible to determine the speed at which the engine would start with any accuracy by any of the methods mentioned earlier. The two-cycle engine, being less stable in operation than the four-cycle, particularly at lower speeds, does not start with any definiteness. Instead, it gradually takes over the load of driving itself as its speed is increased. It was noticed that the required starting speeds for the two-cycle engine were in a region considerably higher than that for the four-cycle. This is as should be expected, however, owing to the higher operating speed of the two-cycle.

Another phenomenon encountered in the cold starting tests as the size of the choking orifices used was decreased was that the engine would start readily with the given size, but then would not be able to continue running because of flooding. From this it can be seen that for an application of a miniature engine requiring ease of starting, a method of automatic control must be provided whereby the amount of choking is decreased immediately after the engine starts.

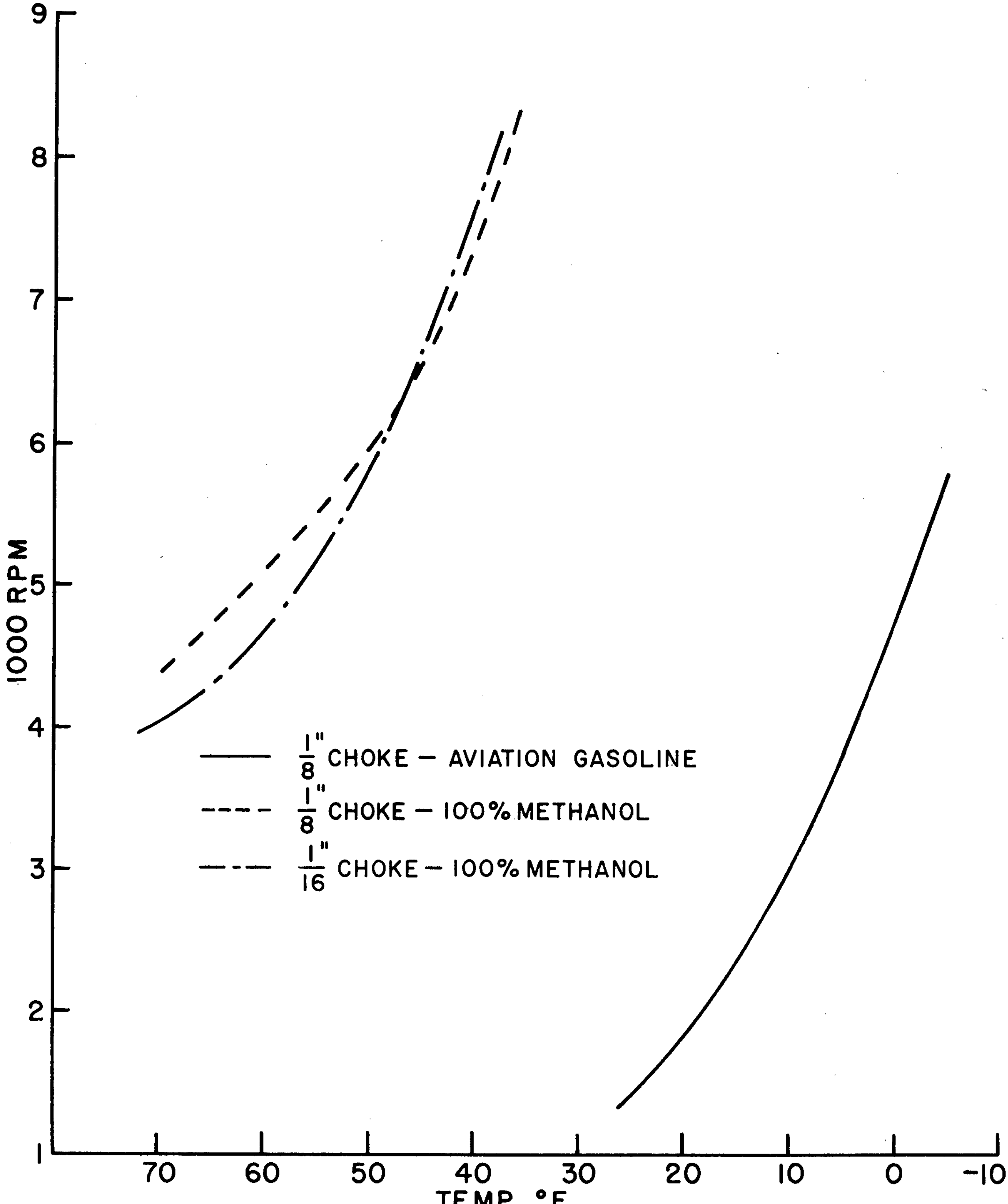


Figure 5. Use of Methyl Alcohol as a Fuel Compared with that of Aviation Gasoline

Experience acquired in cold starting tests using the L-head, four-cycle engine has indicated that a problem can arise due to the increase in viscosity of the lubricating oil. It was found that oil commonly used in the crankcase for testing at normal temperatures, SAE No. 20, became so thick as to prevent the 1/3 hp electric motor from turning the engine over at cold temperatures. As a consequence, lighter oils had to be used for these tests. The use of constant viscosity oils would also solve this problem.

2. STARTING SYSTEMS

The problem of starting internal combustion engines resolves itself into two phases: (1) turning the engine over at a sufficient speed for enough revolutions so as to allow the combustible intake mixture to ignite and take over the job of driving the engine and (2) warming up the engine to a point where it will operate satisfactorily on its normal operating fuel under normal operating conditions.

2.1 THEORETICAL STARTING REQUIREMENTS

In order that the first phase mentioned above is accomplished, stored energy must be available in some form to turn the engine over. In particular, the storage system must be able to release its energy in such a manner as to satisfy the following three requirements:

(1) sufficient total energy must be stored in the system to rotate the engine the required number of times while accelerating to the required starting speed.

(2) sufficient energy must be available for the first compression stroke of the engine.

(3) sufficient crankshaft torque must be available to overcome the maximum opposing torque during the first compression stroke.

In order to determine the total amount of energy to be stored in the system to satisfy the first requirement, it is necessary to know the required engine speed for starting as well as the rate of acceleration of the engine to the desired speed. Knowing the mass moment of inertia and the final speed of the crankshaft and other rotating parts, the kinetic energy of the system at starting can be calculated. Neither the maximum kinetic energy of the piston and connecting rod nor the energy to compress the mixture in the combustion chamber should be added to the total energy requirements since in a complete revolution of the engine the total net torque required by these elements is zero. The number of revolutions needed to reach the desired speed must be known, however, since work is done during every revolution of the engine in overcoming friction. Thus, in general, the total energy requirements of the starting system may be found by adding the kinetic energy of the rotating parts at the starting speed and the total work done by frictional forces in reaching that speed. It may also be necessary to add other work outputs such as the generator and the cooling fan in certain cases.

The second requirement of a starting system to turn the engine over demands that sufficient energy is available during the first half-revolution of the engine so as to do the necessary work of the first compression stroke.

After the top dead center position of the piston is reached, virtually all of this work is recovered on an expansion stroke, possible supplemented by a partial burning of the mixture in the cylinder. Thereafter, it can be assumed that the inertia of the rotating parts will supplement the driving force sufficiently to overcome compression.

The total compression energy required can be expressed by the following equation.

$$E = \int_a^b P \, dV$$

where

E = total compression energy
a = position of piston at beginning of compression
b = position of piston at top of stroke
P = instantaneous pressure
V = volume.

Making the substitution $P = C/V^k$ for an adiabatic compression and integrating, the following equation results:

$$E = \frac{P_a V_a{}^k}{1-k} \left(V_b{}^{1-k} - V_a{}^{1-k}\right),$$

or in other terms

$$E = \frac{P_a \ c \ s \ \pi D^2}{4(c-1)(1-k)} \left(c^{k-1} - 1\right) ,$$

where

P_a = pressure in cylinder at beginning of compression, usually nearly atmospheric, psia
c = compression ratio, based on time of port closing
s = stroke of piston, in.
D = bore of cylinder, in.
k = specific heat ratio (1.4 for air)
π = numerical constant (3.1416).

To determine the value of E as a function of the compression ratio, the following substitution is made:

$$K_1 = \frac{P_a \ s \ \pi D^2}{4(1-k)} .$$

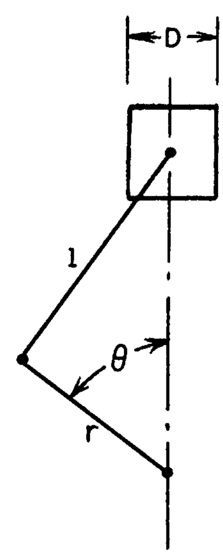
The equation then becomes:

$$E = \frac{K_1 \, c(c^{k-1} - 1)}{c-1} .$$

Values of E/K_1 have been plotted as a function of compression ratio in Figure 6. It can be seen that a doubling of the compression ratio from the value of 6 to 12 results in an energy increase of 48%. Furthermore the curve is nearly linear in this range.

In addition, the frictional work for the length of the first stroke must be added.

In order that the third requirement of starting be satisfied, the system must have sufficient crankshaft torque available to overcome the maximum opposing torque during the first compression stroke. If we let θ equal the angle between the crank arm and the centerline of the piston as shown below, it can be seen that the maximum torque will invariably occur at an angle θ which is considerably less than 90°. This is due to the fact that the cylinder pressure increases at a faster rate than the change in effective crank arm length in this vicinity.



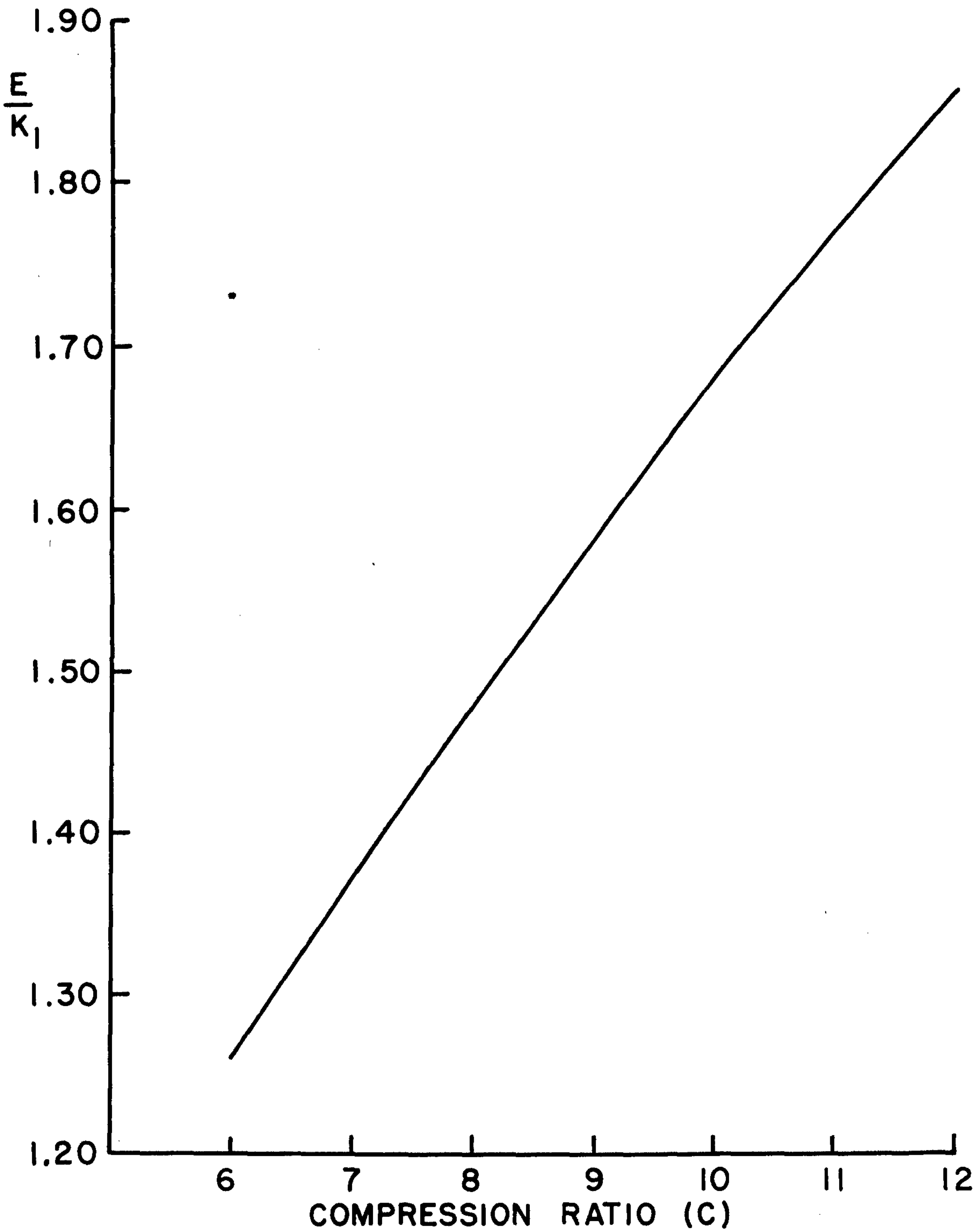


Figure 6. Energy of First Compression Stroke as a Function of Compression Ratio

Assuming an adiabatic compression, the equation for the torque on the crankshaft due to the compressive force acting upon the piston can be expressed as:

$$T = P_a \, s \, \frac{\pi D^2}{8} \sin\theta \left[\frac{2c}{2c - (c-1)\cos\theta}\right]^k \left[1 + \frac{\cos\theta}{(l/r)^2 - \sin^2\theta}\right]$$

where

- T = torque on crank shaft, in.lb
- θ = angle between crank arm and centerline of piston
- P_a = pressure in cylinder at beginning of compression, usually nearly atmospheric, psia
- s = stroke of piston, in.
- π = numerical constant (3.1416)
- D = bore of cylinder, in.
- c = compression ratio
- k = specific heat ratio (1.4 for air)
- l = length of connecting rod, in.
- r = length of crank arm, in.

This equation assumes that compression begins at the bottom dead center position of the piston and is only approximate in cases such as the two-cycle engine where compression begins after the ports are closed. This error is usually not of high magnitude, however.

For a given case, the constant K_2 may be used as follows:

$$K_2 = P_a \frac{\pi D^2 s}{8} \; .$$

The equation then becomes:

$$T = K_2 \sin\theta \left[\frac{2c}{2c - (c-1)\cos\theta}\right]^k \left[1 + \frac{\cos\theta}{(l/r)^2 - \sin^2\theta}\right]$$

Differentiating T with respect to θ to find the maximum point results in an equation which cannot be solved explicitly for θ; therefore the equation has been solved numerically for maximum values of the dimensionless ratio T/K_2 for the various values of c and l/r. These results have been plotted on the curves in Figure 7 assuming $k = 1.4$. Values of l/r are usually in the range of 3 to 4 for most miniature engines; however, the L-head, four-cycle engine has an l/r ratio of 5.17.

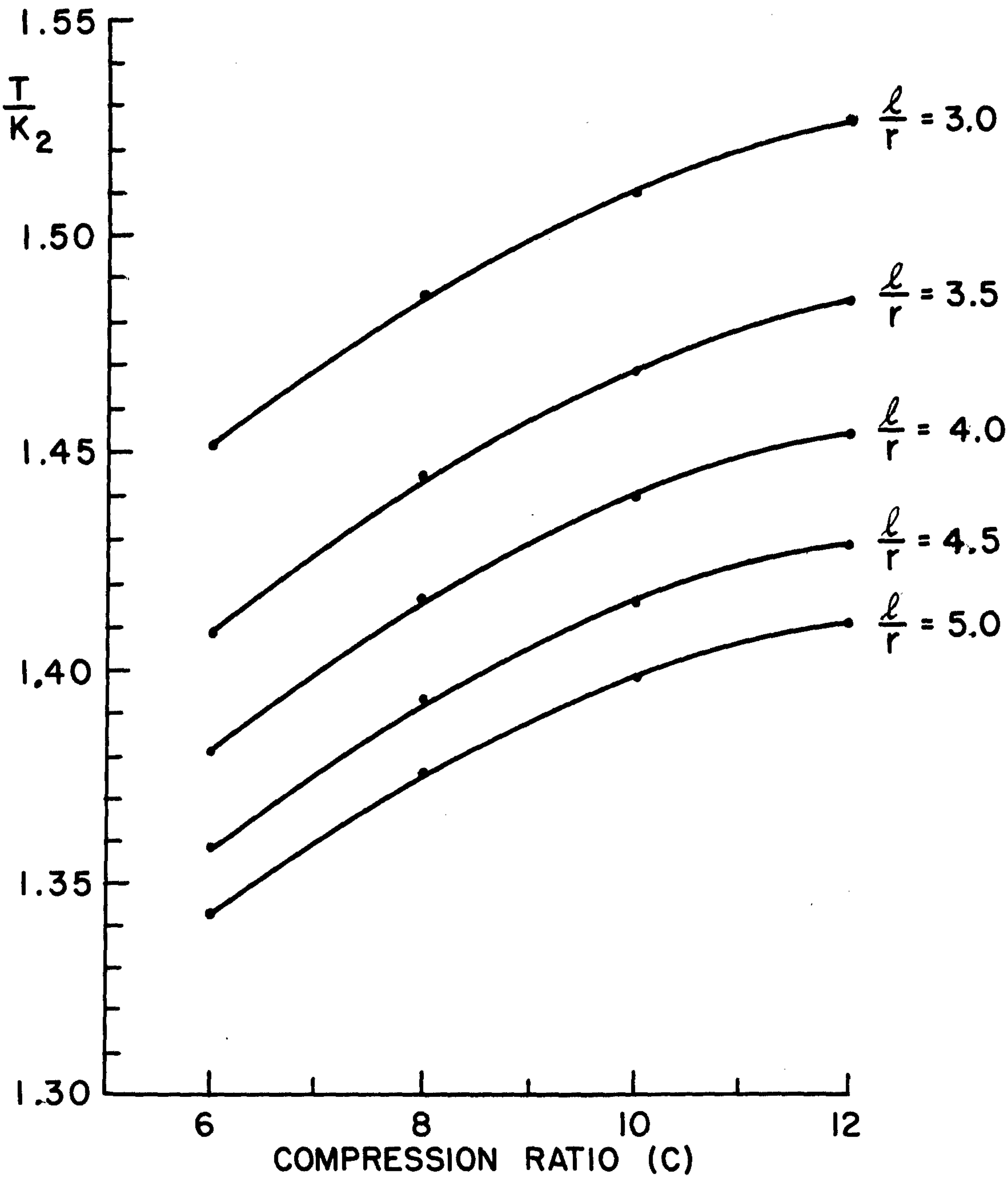


Figure 7. Compression Torque as a Function of Compression Ratio and ℓ/r Ratio

It is seen that the ratio T/K_2 increases as the compression ratio increases or the l/r ratio decreases, as one might expect. However, the change in torque in this range of compression ratios is not great. For instance, for the l/r ratio of 4, a doubling of the compression ratio from 6 to 12 results in only a 5% increase in maximum torque.

In addition, the torque due to friction must be added to this for the total amount of opposing torque to be overcome by the starting system.

Thus, in the design of any starting system for miniature engine-generator sets, all three requirements discussed above must be satisfied in order for starting to take place.

2.2 ENGINE WARM-UP

The latter phase of this problem involves a proper selection of fuel for the warm-up period depending upon environmental operating demands. For example, starting at low temperatures requires a volatile fuel until the intake manifolds are warmed sufficiently to vaporize one which is less volatile. This requirement may be met by means of a dual-fuel starting system utilizing one fuel for warming up and another for normal operation. Such a system was discussed in the previous section concerning low temperature starting when alcohol was used. Typical fuels to be used for starting might be hydrogen, acetylene, propane, and butane. Thereafter, the system could convert to a different fuel for operation, such as gasoline, alcohol, or kerosene.

In a dual-fuel system such as the one described, there are various methods which could be used to store the two fuels. The fuels could be stored separately in their own respective containers which might result in a low pressure container for the operating fuel and a high pressure container or a gas generating system for the warm-up fuel. Or it might be possible to store both fuels in one container; e.g., hydrogen and gasoline in one container at a moderately high pressure. This latter method would require two fuels which would not interact under the expected temperatures and pressures.

In any event, regardless of the method of storage, a dual-fuel supply would probably require two fuel-metering systems. However, if it were possible to use a fuel which would be satisfactory for both starting and operation, the over-all reliability of the system would be increased because of the decrease in its complexity. A fuel which might satisfy these conditions is propane. It has a heating value in Btu/lb which compares favorably with gasoline and other paraffin fuels and it has a boiling temperature of -44°F at atmospheric pressure. Also, the latent heat of vaporization of propane is nearly the same as that of gasoline. Because of the temperature gradient between the environment and the fuel, it can be seen that propane would vaporize under all possible operating conditions involving low temperatures.

The Ruckstell-Hayward Beacon engine-generator unit employs hydrogen as an auxiliary fuel for warm-up purposes. This fuel was used to avoid carburetion problems for the auxiliary fuel, particularly under cold starting conditions. Hydrogen has a wide range of inflammable mixtures (from 4% to 74% by volume of fuel vapor in mixture); therefore, it was felt that the engine would operate easily with this fuel. Nevertheless, extensive tests with this unit showed considerable difficulty in operating the engine with hydrogen, at every mixture ratio. It is felt that the difficulty is largely due to the high ignition temperature required by hydrogen.

2.3 STARTING SYSTEMS

Starting systems can be classified into two groups - manual and automatic. Basically, the manual system assumes the presence of a human being to act as a prime mover to crank the engine over and possible also to make certain adjustments after the engine is running. The automatic system requires no person present to start and operate. The energy required to crank the engine over is stored in a storage device and all controlling is done automatically. An automatic-starting, miniature engine-generator set could be developed so as to start upon reception of a radio signal or other impulse. Generally, all the starting systems described below require some sort of clutching device to allow the engine to disengage after starting.

2.3.1 Manual Starting Systems

A manual starting system must provide for the inherent physical limitations of the human being; i.e., the system must not require above-average strength, speed, and dexterity to operate. With these limitations in mind, the following three types of manual starting systems have been analyzed:

(1) manual rope pull starter
(2) hand cranked inertia starter
(3) hydraulic cranking motor

Block diagrams of each of the above systems appear in Figure 8.

The manual rope pull starter has been used successfully for many commercial applications employing small engines, such as outboard motors, chain saws, lawn mowers, etc. This method of starting is highly advantageous for a system requiring light-weight, low-volume characteristics. It cannot be used for engines requiring a combination of high starting torque and speed, however. Also, starting at colder temperatures may be difficult due to the higher starting speed required, in addition to the higher frictional torque present.

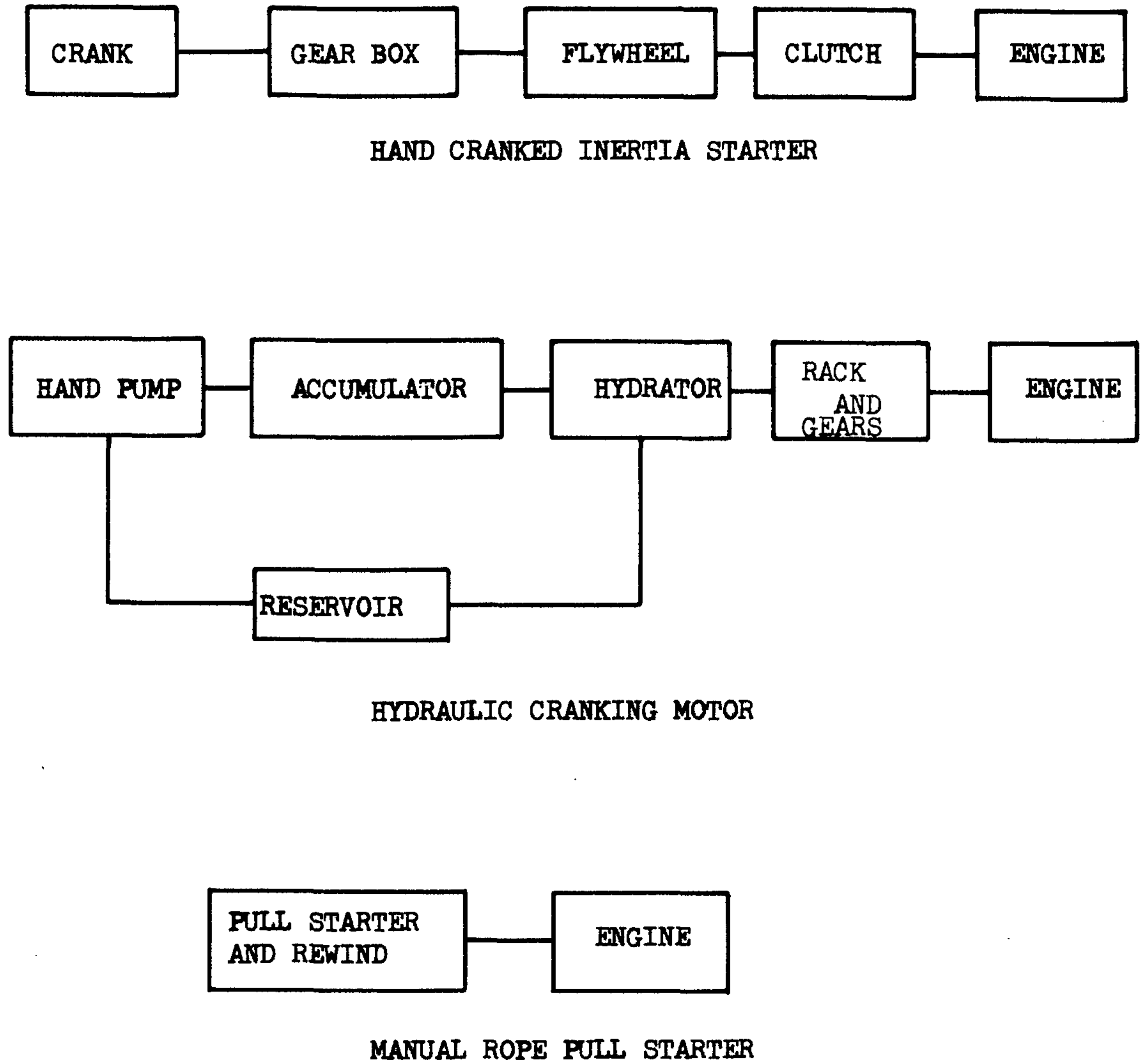


Figure 8. Block Diagrams for Manual Starting Systems

The hand cranked inertia starter would be a good system to use for an engine requiring a high starting speed and fairly high starting torque. With this system, a flywheel would be gradually accelerated to a high speed by means of an external hand crank and a step-up gear set with the engine being disconnected. Then after a sufficiently high flywheel speed is reached, a clutching device could be used to engage with the engine to turn it over. A miniature engine-generator set may have sufficient rotary inertia in the generator armature to act as the flywheel and may thus avoid the increased weight and volume of an additional flywheel.

The hydraulic cranking motor, as described in Figure 8, has a potentially high capacity of energy storage for use in starting situations requiring high speed and/or high torque. This system, however, would require a much greater weight and volume, which would eliminate it as a possibility for many applications. The system employs a hydraulic hand pump to pump a fluid through a check valve into an accumulator. The accumulator consists of a piston and cylinder. On one side of the piston is a compressed gas at a fairly high pressure, possibly in the neighborhood of 500 psi, and on the other side is the hydraulic fluid. As fluid is pumped into the accumulator, energy is stored by means of increasing the pressure on the compressed gas. When sufficient energy has been stored in the accumulator, a rapid-opening control valve is opened, which allows the hydraulic pressure to act upon a piston in the hydrator which, in turn, drives the engine by means of a rack and a step-up gear system. After starting, the opening of another valve allows hydraulic fluid to flow from the hydrator to a reservoir where it is stored for future engine startings.

2.3.2 Automatic Starting Systems

The following types of automatic starting systems have been analyzed and are illustrated in Figure 9:

(1) cartridge starter
(2) electrical cranking motor
(3) electrically cranked inertia starter
(4) compressed gas starter
(5) spring starting system.

The cartridge starter shown in Figure 9 employs a combustion chamber in which a cartridge containing mechanite or another material is inserted. The high pressure gas which then results from the rapid burning of the cartridge is used to crank the engine by means of a piston, rack, and step-up gear system.

Cartridges are widely used for the starting of larger internal combustion engines. For example, one starter was developed for aircraft engines near the beginning of World War II, which had a total weight of 37 lb and was used to start 1,800 hp engines (Ref. 18). This system employed a breech to fire a cartridge similar in appearance to a shotgun shell. These cartridges came in several sizes to be used for different

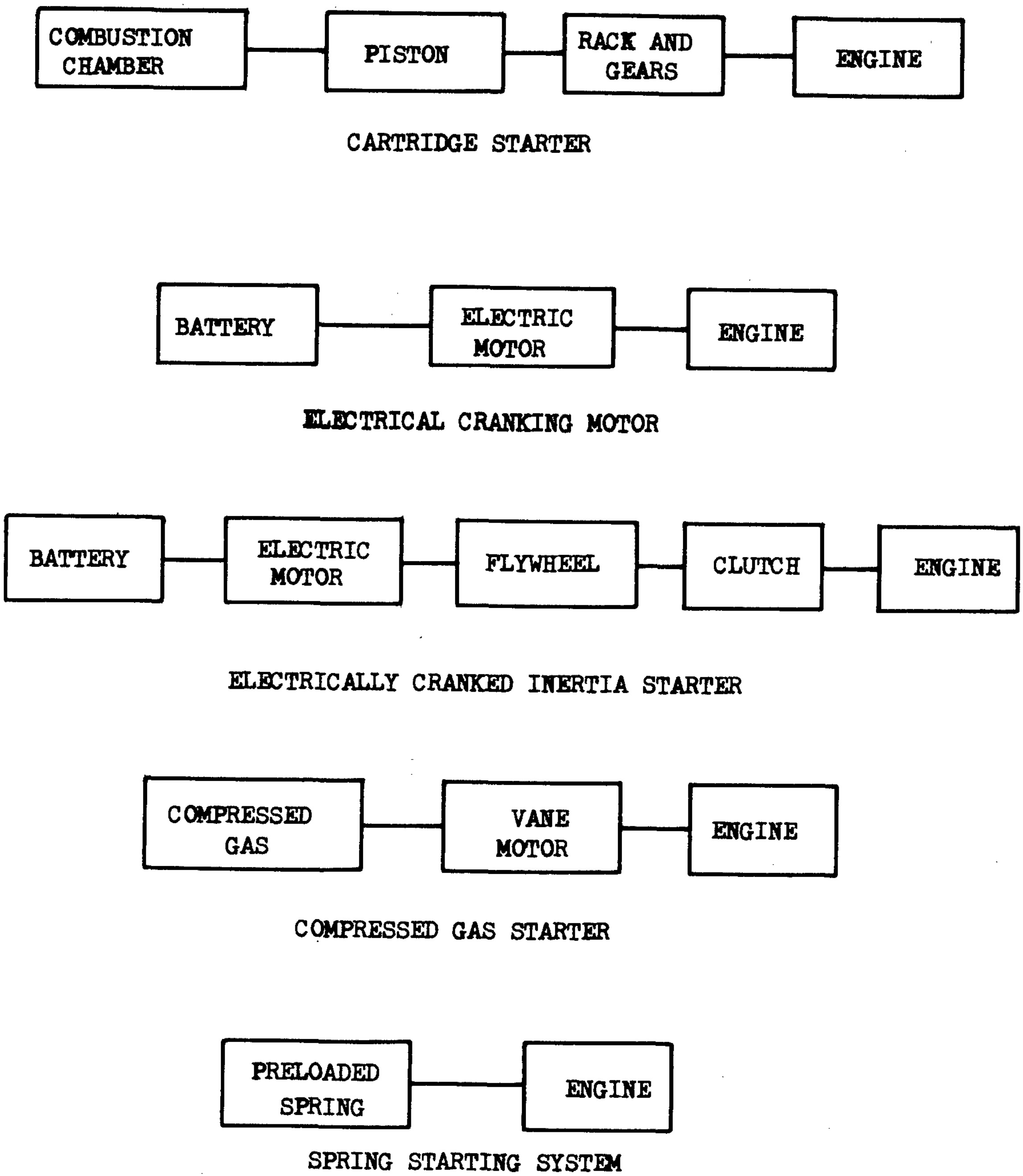


Figure 9. Block Diagrams for Automatic Starting Systems

applications and contained a mixture with relatively slow burning characteristics similar to those of commercial cellulose. A steel connecting tube ran from the breech to a cylinder containing a piston which, in turn, drove the engine crankshaft by means of a speed multiplier.

To avoid the necessity of having the extra weight and volume of an auxiliary cylinder and piston and a step-up gear system, another method which eliminates these components may be feasible. This method employs firing from the breech directly into the engine cylinder with piston slightly beyond top dead center position. This may be practical for engines with low starting speeds, since the engine must be accelerated to the necessary speed in one-half revolution of the crankshaft. It was found to be successful in starting tractor engines in Great Britain.

The explosive charge used in the cartridges undoubtedly has a high energy storage capacity on both a weight and volume basis, as well as the ability to release its energy in a very short time. Its reliability may be somewhat questionable, however, depending upon the reliability with which burning characteristics of the cartridges may be predicted.

An automatic starting system may employ an electric cranking motor to turn the engine over. Such a motor must have sufficient torque to overcome the initial compression. Electric dc motors operating from low voltage batteries are available. The intermittent operation to which such a motor would be exposed would allow for a compact unit to be used. If the particular miniature engine-generator set has a dc generator, the design may be changed to allow the generator to act as a motor for starting purposes. High voltage batteries may be used to supply current for this purpose.

Another type of starting system which uses an electric motor is the electrically cranked inertia starter. For this system, a small, high-speed, dc electric motor may be used to accelerate a flywheel without the engine being connected. This system is the same as the hand cranked inertia starter discussed earlier in this section except for the difference in prime movers.

A study of energy storage devices was made as part of a doctoral dissertation at the Ohio State University published in 1954. The results of this study are shown in Tables A and B. These tables represent energy storage capacities of various systems on a weight and volume basis. These values represent realistic rather than optimum figures. For example, in every case except the electrochemical cells, the material being used as a storage medium is steel with a yield point of 100,000 psi.

It can be seen from the tables that the compressed gas storage system ranks fairly high on both a weight and volume basis. The system is also able to release its energy fairly rapidly. Thus, a compressed gas starter may be used to start an engine; such a system may employ gas stored in a tank under high pressure or may use gas which is chemically generated when needed. Gas generation may eliminate leakage problems; however, problems

TABLE A. ENERGY STORAGE CAPACITY PER UNIT WEIGHT

System	ft lb /lb
Ag-Zn electrochemical cell	146,000
Lead-acid electrochemical cell	38,000
Edison electrochemical cell	35,000
Uniform stress flywheel with rim	27,350
Compressed gas (spherical container)	22,600
Compressed gas (cylindrical container)	18,400
Cylindrical flywheel	18,000
Rib-spoke flywheel	2,450
Compressed liquid (ether)	300
Compressed solid (torsion spring)	29
Compressed solid (spiral-wound spring)	16
Compressed solid (helical coil spring)	15
Compressed solid (Belleville spring)	11

TABLE B. ENERGY STORAGE CAPACITY PER UNIT VOLUME

System	(ft lb /ft^3) x 10^{-3}
Ag-Zn electrochemical cell	14,800
Uniform stress flywheel with rim	9,000
Cylindrical flywheel	8,750
Compressed gas (spherical container)	5,200
Lead-acid electrochemical cell	5,000
Edison electrochemical cell	4,800
Compressed gas (cylindrical container)	4,000
Rib-spoke flywheel	1,440
Compressed liquid (ether)	90
Compressed solid (torsion spring)	8
Compressed solid (helical coil spring)	5
Compressed solid (spiral-wound spring)	4
Compressed solid (Belleville spring)	2

in triggering the starting mechanism are introduced. The compressed gas may be used to drive a vane motor or a piston which, in turn, drive the engine.

The Ruckstell-Hayward Beacon starting system received extensive testing. This unit employed nitrogen stored at 2,000 psi to drive a vane motor. The nitrogen pressure was regulated to 160 psi before entering the vane motor. It was found that this starter was very successful for the purpose for which it was intended. After many starting attempts, the graphite blades of the vane motor were found to wear excessively; however, they were replaced with ones made of formica which were very adequate.

The last type of starting system listed is the spring system. Such a system may employ a clock-type, spiral-wound spring which is wound about the crankshaft. This sytem would be compact and extremely reliable.

3. ENGINE-GENERATOR COOLING

INTRODUCTION

The thermal considerations of an engine-generator set should be one of the major considerations of the designer. The degree of success in the operation of a power generating unit is directly related to the effectiveness of the methods of heat removal.

The analytical solution of the engineering problem of cooling engine-generator sets is a very difficult one because of the complexity of the sequence of events. The engine gas temperatures and velocities pass through a complex cycle. Also the heat transfer within the cylinder occurs by the processes of radiation, convection and conduction.

Because of the capacity of engine and generator parts to store heat, their temperatures do not follow the instantaneous variation in heat flow during the cycle but tend more nearly to attain time average values. The experimental data presented in the following paragraphs was obtained with the engine and generator at thermal equilibrium. All the temperatures recorded were time average values.

3.1 COOLING SYSTEMS

In the design of a cooling system for an engine-generator set, four limitations are usually imposed: 1) cylinder-head temperatures must be maintained at a value which will assure reliable operation and engine life; 2) the energy required to cool the set should be a small percentage of the total energy released; 3) the cooling system must not increase the total weight of the equipment above a specific amount; 4) the cooling apparatus should be so arranged as not to appreciably change the total volume of the set.

3.1.1 Air-Cooled Engines

Because of the high rates of heat dissipation per unit area of cylinder surface, free or natural convective cooling cannot be used without attaining prohibitive cylinder temperatures. Forced air cooling is required to assure reliable operation. Increasing the air velocity past the heated cylinder results in a decreased resistance to heat transfer across the air film and in an increase in cooling which will be much greater than for free convection.

A good design of an air-cooled engine which incorporates simplicity, reliability and compactness is (Figure 10) the Ruckstell-Hayward "Pockette." Proper engine operating temperature is maintained by a controlled flow of cooling air past the finned cylinder head. The cooling air fan is mounted on the power plant shaft and contains radial, backward-curved blades. Air is taken into openings in the starting rotor housing,

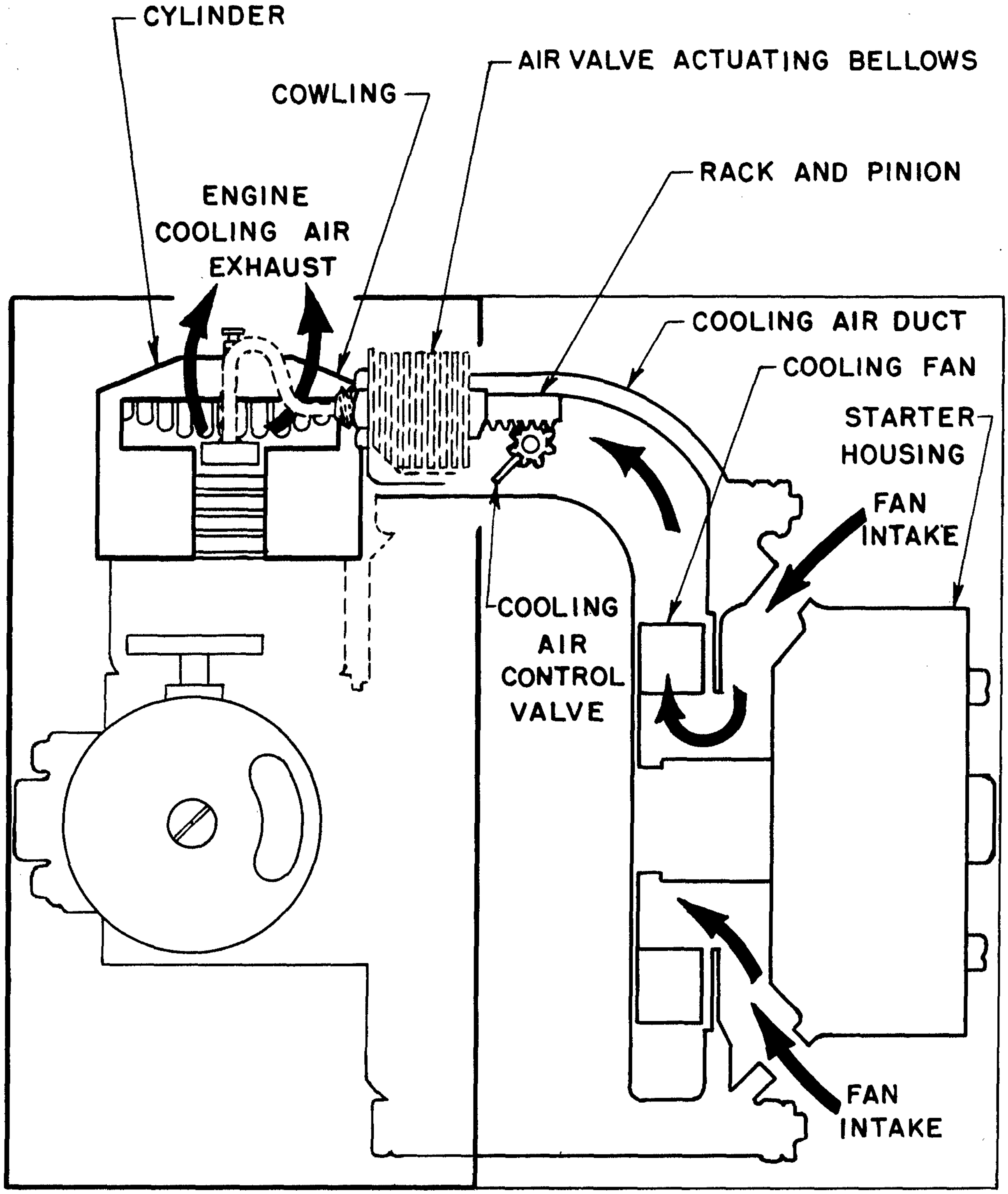


Figure 10. Cooling System Diagram. Ruckstell-Hayward "Pockette"

forced by the cooling fan past a thermally-controlled valve in the air duct in the generator and fan housing, through a cowling covering the cylinder head, and ducted overboard through a screened opening in the back panel.

The valve in the cooling air duct is linked by a rack-and-pinion gear system to a bellows which controls the rack movement. The bellows is thermally controlled, acting in response to cylinder head temperatures through connection to an aniline-filled bulb, which is integral with the cylinder head.

3.2 BLOWERS

The purpose of this section is not to discuss any specific phase of blower design to a great extent, but rather to discuss the different types of blowers and their range of application. Detailed information for blower selection can be found in company catalogs or technical literature. Three sources applicable to the selection of a proper fan or blower are presented in references 17, 34 and 39. Several companies have developed a complete line of fans and blowers which cover the operating range required for engine-generator applications. These miniature fans and blowers are used extensively for cooling of airborne, electronic equipment and thus are readily available commercially. It is recommended that a designer of cooling systems for miniature engine-generator sets select a fan or blower that is already developed, or at least pattern the impeller and scroll design after one of the types which have proven satisfactory for the ratings desired.

Both narrow, straight-bladed and multiblade, forward-curved centrifugal fans have been used for cooling small and miniature engines. The forward-curved blade design gives the smallest size of blower for a given capacity. For applications where the specific speed is between 3,000 to 20,000, the narrow straight-bladed centrifugal blower may be used to advantage (reference 17). For specific speeds of 20,000 to 50,000, the forward-curved blade design may be the most satisfactory, while for specific speeds in excess of 50,000 the vane-axial fan may be the best design. It is pointed out that these ranges are useful as a rough guide only, and that there is considerable overlap in all ranges. However, they do give the designer some indication as to the types of blowers and fans which should be considered.

The term specific speed is used to classify fans and blowers on the basis of their performance and proportions regardless of their actual size or the speed at which they operate. The relationship between volume flow rate Q, in cubic feet per minute, P_s, the head or total static pressure in inches of water, N, the rotational speed in revolutions per minute and the specific speed N_s is expressed by the following equation.

$$N_s = \frac{N\,Q}{{P_s}^{3/4}} \quad .$$

For miniature engine-generator cooling it appears that specific speed values of 6,000 to 60,000 might be encountered in different types of equipment, with 25,000 a typical value for a set to provide 100 to 200 watts output. Thus, it appears that centrifugal blowers with forward-curved blades might well be the optimum selection. However, the designer should make a careful comparison between the vane-axial type of design and the centrifugal.

Some empirical formulae which give a ready indication of the size and performance of present designs of small forward-curved blade centrifugal blowers are presented (reference 39 and nomograph of reference 17). These formulas are based on air at standard density of 0.075 lb/ft^3 and for operation at maximum blower efficiency.

$$\text{Impeller diameter, in.} = \frac{12500\ P_s\ (\text{static pressure, in. water})}{N\ (\text{rpm})}$$

$$\text{Compressor scroll diameter} = \frac{0.8\ Q(\text{cfm})\ N(\text{rpm})}{P_s\ (\text{in. water static})\ 10^5} \quad .$$

The approximate volume of small centrifugal blowers (without motor) may be estimated from the equation:

$$V(\text{volume, in.}^3) = 3.5\ w\ (d_1)^2$$

where

w = axial width of blower impeller, in.
d_1 = impeller diameter, in.

$$W(\text{weight, lbs}) = 1.28\ \frac{V(\text{in}^3)}{100}^{0.8} \quad .$$

The above equations for weight and volume of small centrifugal blowers are only approximate due to variation in design among manufacturers.

3.2.1 Axial-Flow Blowers

In the axial-flow blower, the path of the air stream is parallel to the drive motor and impeller. It depends on the conversion of kinetic energy into air pressure. There are several manufacturers that carry a complete line of aluminum and magnesium blowers. They can be purchased as small as 10 ounces in weight with a 2-inch fan diameter. The space saving compactness of vane-axial designs allows installations in small space allotments. No more space is needed than that required by the duct system. Vane-axial design also provides high capacity performance with low horsepower expenditure. However, the vane-axial blower has poor control

characteristics and must assume complicated multi-stage configurations in order to be capable of appreciable air pressure production. In installations requiring air only at low pressures, single-stage blowers, the vane-axial is comparable to the centrifugal.

3.2.2 Design of a Centrifugal Blower

Performance curves for a small centrifugal blower are presented in Figure 11. This blower was used on the two-cycle engine-generator set designated as the Mark III. The blower design is nearly identical to that employed to cool the Ruckstell-Hayward Pockette engine-generator sets. The impeller is 2.5 inches in diameter and 0.36 inch wide.

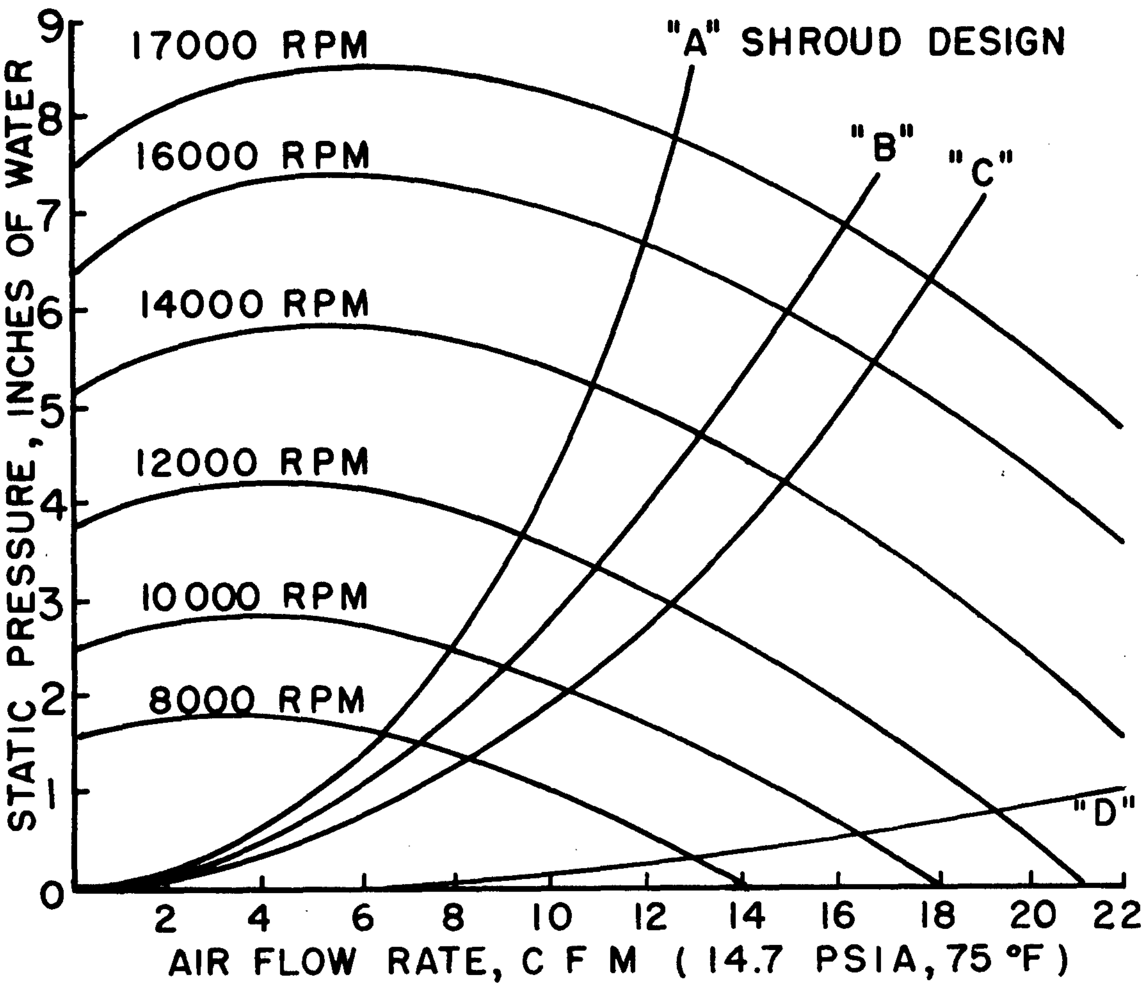


Figure 11. Performance Curves for Radial-Blade, Centrifugal Blower and System Resistance Curves for Different Designs of Engine Cooling Shrouds

The blades are radial at the tip, but are curved forward at the root to reduce shock at inlet and improve blower performance. The pressure rise is somewhat less than a blower with forward-curved blades, as indicated by the preceding equations. However, the width of the impeller agrees with the above equation for impeller width perfectly, if it is calculated at points which should give about maximum efficiency for this type of blower.

Ex: 16,000 rpm, 517 inches water, 16 cfm)

Test results indicate the efficiency was not as good as that claimed for miniature blowers of this size. The general performance of the blower is typical of blowers with this type of blade design. Also shown in Figure 11 are the system resistance curves of four different shroud designs for this engine. As discussed in the following section, a considerable increase in the flow rate can be achieved for equal pressure drops simply by careful attention to the shroud design.

3.2.3 Fin and Shroud Design

In the design of an air-cooled engine, the designer should strive to obtain maximum heat transfer with a minimum expenditure of cooling power. This optimum condition can be obtained by proper finning and baffling. The solution to this problem is further complicated by the non-isothermal surface of the engine. The temperature difference between intake and exhaust ports can be appreciable. Also the cylinder head which is exposed to the products of combustion for a longer period of time than the cylinder wall is a potential "hot spot." To assure reliable engine operation, these "hot spots" must be eliminated, and the average cylinder temperature must be held below a specified maximum. Such a condition may be controlled by proper baffling and finning, neither of which takes the place of the other. Finning is used to give the cylinder a large heat transfer coefficient, and baffling to increase the air flow at a given pressure drop and to control the temperature distribution. Both methods have certain limitations. Fin depth is limited by the inefficiency of the fin tip for deep fins. The effectiveness of baffles or shrouds is limited by the available pressure drop and by entrance and exit losses that cannot be totally eliminated.

The engineering problem of correct fin design is not only one of determining the geometry of the fins such as thickness, width, and space between adjacent fins for maximum heat transfer, but it also involves the factors imposed by manufacturing and service requirements such as strength, machinability, and weight. The designer is faced with the difficult task of selecting the proper fin dimensions to satisfy the most important requirements. Two very useful sources of information that can be used as a guide in correct fin design are found in references 6 and 29.

Three different types of shrouds that are commonly used is illustrated in Figure 12. Shroud (a) approaches the ideal, in that maximum heat transfer is obtained with minimum cost of cooling power. The air entering the shroud is split into two streams which reunite at the exit.

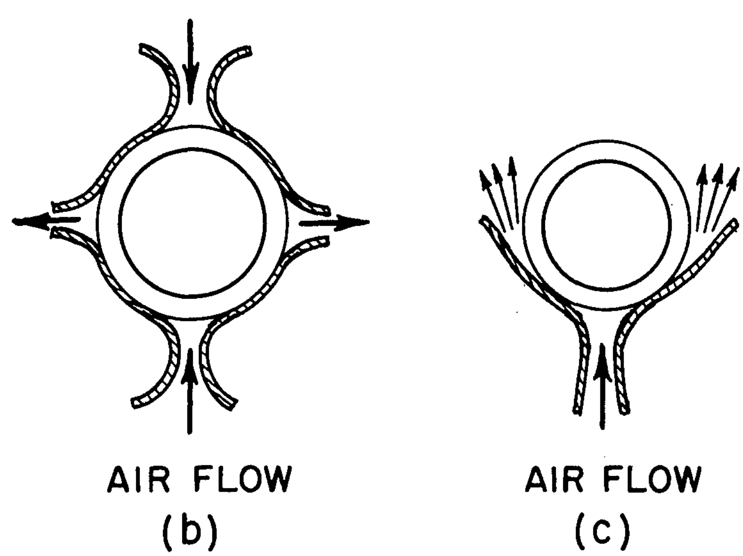


Figure 12. Shroud Designs for an Air-Cooled Engine

There is a loss in total pressure head due to the turbulence created. In Figure 12 (c) the air is in contact with only about one fifth of the cylinder. The pressure drop is less than (a), however, the rear of the cylinder is not cooled resulting in a potential "hot spot." The two air streams must come together from opposite directions to assure proper cooling of the entire cylinder. In Figure 12 (b), the length-diameter ratio of the air passage is half the value of that obtained in 12 (a). If the pressure drop across the cylinder is held at the same value as 12 (a), the fins can be spaced twice as close together. The result is twice as much heat transfer surface and twice as much cooling. However, because of the increased space requirements and ducting problems this shroud design may not be feasible.

3.3 GENERATOR COOLING REQUIREMENTS

The generator cooling requirements are relatively small in comparison to the internal combustion engine. About 20 per cent of the total heat load comes from the generator. The only practical way to cool the stator and

and rotor is by forced-air cooling. Generally, the blower is mounted on the same shaft as the rotor. The generator is cooled by induced air entering the blower housing. The compressed air leaving the blower scroll is directed into the shroud surrounding the engine. If the total air required for the cooling system passes through the generator, it will operate over-cooled, which imposes an unnecessary penalty upon the system. Thus, air ports should be provided downstream allowing the major portion of the cooling air to by-pass the generator. Cooling power requirements for the generator can be further reduced by more efficient electrical design, use of materials having higher temperature limits, and balanced thermal design.

A study of air-cooled aircraft generators has been in progress for several years in the Mechanical Engineering Department of the Ohio State University. Several reports have been published which contain information that should be very valuable to the electrical engineer in the design of miniature generators.

3.4 FORCED LIQUID COOLING

With forced liquid cooling, the heat transferred per unit area is greater than with a natural or free convective liquid cooling system. The energy required to force the liquid through the system is converted into heat which increases the temperature of the coolant. The total heat rejected at the ultimate source is increased, therefore, by the value of the pumping power However, it is generally a small percentage of the total heat dissipated and can be neglected when considering the resulting improvement in rate of removal of generated heat at the engine. The system packaging becomes more involved with the addition of a liquid pump and piping. There are several manufacturers that carry a complete line of miniature pumps suitable for small engine-generator sets in the range considered in this report. Small gear type pumps have the advantage of not vapor locking as readily as centrifugal pumps. Furthermore, gear pumps have the advantage of being smaller in size and weight with higher operating efficiencies.

The packaging of a liquid cooled motor-generator set is further complicated by the addition of a heat exchanger which transfers the heat conveyed by the coolant to an ultimate sink. This heat exchanger may consist of finned tubes cooled by forced air or a system using an expendable evaporative coolant as shown in Figure 13. One method that may be used is to build the heat exchanger into the shell housing the motor-generator set. In this case, the heat would be dissipated to the surrounding air by free convection. The shell temperature and thus the cylinder temperature would be dependent on the ambient air temperature.

3.4.1 Vaporization Cooling

In vaporization cooling systems, heat is removed from the cylinder by the vaporization of the coolant from the fluid state into the vapor s

The amount of heat removed per pound of coolant is dependent upon the latent heat of vaporization. Its value depends on the saturation pressure or temperature and decreases as the pressure increases.

The simplest vaporization cooling system consists of a jacket surrounding the cylinder where heat is transferred directly from cylinder to the boiling liquid. If the vaporized coolant is expended through a valve to the atmosphere, expendable evaporative cooling, the cylinder temperature i a function of the vapor pressure in the boiling chamber. When the tempera ture difference between a boiling liquid and the cylinder wall is increased, the rate of boiling increases; therefore, more heat per unit area is removed. As this temperature difference increases, a maximum critical temperature difference is reached beyond which the rate of boiling decreases. If the temperature difference becomes greater than this critical value, the vapor formed by boiling acts as an insulator, impeding the transfer of heat.

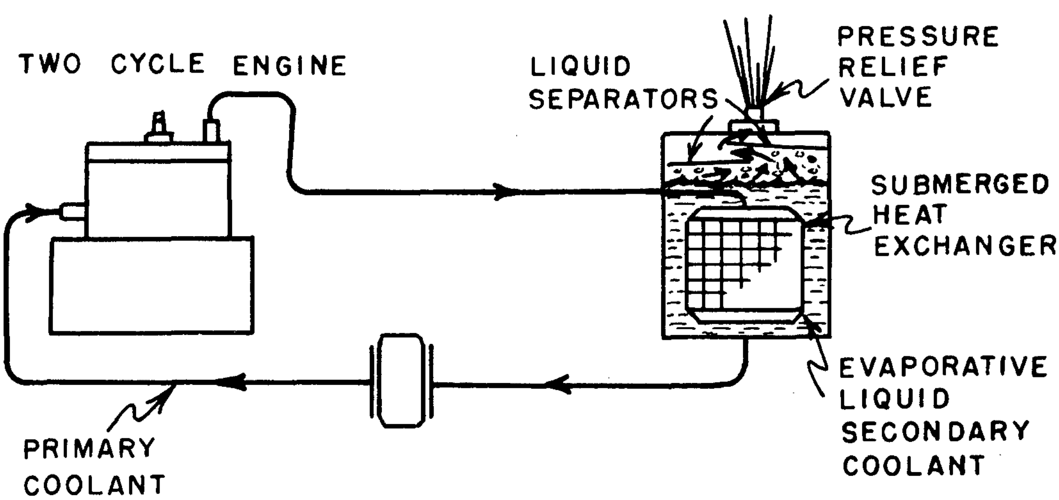


Figure 13. Schematic Diagram of Expendable Evaporative Cooling System

In the cooling system shown schematically in Figure 13, the heat dissipated by the engine is transferred by a primary coolant to a heat exchanger submerged in a secondary coolant. The primary coolant remains a liquid throughout its cycle. This type of system is preferable over the one mentioned above because of fewer mechanical design problems. Also due to the larger heat transfer area available in the submerged heat exchanger, the ebullition is not as violent. Thus, the problem of separation of entrained liquid in the vapor is simplified.

The operating period of an evaporative cooling system is determined by the rate of heat dissipation, the latent heat of vaporization of the coolant and the quantity of coolant available. Calculations indicate that for small engine-generator sets that are to operate in excess of 10 hours the additional weight of the expendable coolant becomes prohibitively large as compared to an air cooled system.

3.4.2 Coolant Selection

In selecting a coolant for a liquid cooled engine there are several factors that the designer should investigate. It is necessary for the liquid to be chemically and thermally suitable over the entire operating temperature range. The pertinent properties that should be considered are freezing point, vapor pressure, latent heat of vaporization, viscosity, thermal coefficient of expansion, thermal conductivity, toxicity and chemical inertness. It is desirable that a coolant be inert with respect to all materials with which it comes into contact and care must be taken in the choice of such parts as gaskets to prevent leakage.

Water is an ideal heat transfer fluid except for its high freezing point. It is widely used in cooling systems with additives to suppress its freezing point. Ethylene glycol and methyl alcohol are two common additives. An excellent source of information to aid the designer in the selection of a proper coolant for a liquid-cooled system is found in reference 12.

3.5 ENGINE COOLING REQUIREMENTS

3.5.1 Ruckstell-Hayward

To investigate the cooling requirements of air-cooled engines, tests were conducted on three engines. The first engine tested was the Ruckstell-Hayward two-cycle loop-scavenged engine shown in Figure 10. The cylinder and head of the engine are made of steel with a total cooling surface of 26 square inches. The fuel used was 90 per cent alcohol and 10 per cent castor oil. The total heat load to the cooling air had a range of 1300 to 3000 Btu per hour. This heat dissipation to the cooling air was 5 to 25 per cent of the heating value of the fuel. The limits may be slightly in error due to difficulty in measuring the fuel requirements for this particular engine. Cylinder head temperatures ranged from 430 to 550°F.

The cooling air requirements for this engine ranged from 15 to 25 cubic feet per minute. The air flow rate was measured with a flowrotor. The shop air entered a specially constructed air duct attached to the inlet side of the blower (Figure 10). Pressure taps located near the blower inlet allowed the operator to adjust the amount of shop air entering the air duct to equal that being withdrawn by the blower. All temperatures measurements were made with iron-constantan thermocouples.

3.5.2 Mark III

The second engine tested was the two-cycle Mark III engine. The fuel used was 80 per cent alcohol and 20 per cent castor oil. The total cooling surface of the engine is 28 square inches. The amount of cooling air was measured with a flowrotor. Engine surface temperatures were obtained by seven iron-constantan thermocouples which were cemented to the engine with an epoxy resin. Six thermocouples measured the inlet and exit air temperatures. Two different types of air shrouds were used in the test runs. Shroud A was machined from aluminum with very small clearance between shroud and engine cooling fins. Shroud B was constructed from galvanized sheet metal with clearances between shroud and fins ranging from 1/8 to 3/8 of an inch.

The heat dissipated to the cooling air was 5 to 15 per cent of the heating value of the fuel. Figure 14 shows that the cooling load varied from approximately 1600 to 2300 Btu per hour for Shroud A. The corresponding cylinder head temperature associated with Shroud A varied from 340 to 290 $^{\circ}$F, average values. With Shroud B the heat removed from the engine varied from 800 to 1900 Btu per hour with a corresponding average cylinder head temperature of 545 to 455 $^{\circ}$F. The range of cooling air was varied from 10 to 28 cubic feet per minute. The surface heat transfer coefficient of the engine increased approximately as the 0.8 power of the cooling air rate. This is evident by the increase in heat removal with a corresponding decrease in cylinder head temperature with increasing air flow rate. In general, the cooling load increased with engine load. With Shroud B, the data shows an average increase in cooling load of 200 Btu per hour with an increase in engine load from 0.26 bhp to 0.41 bhp. Undoubtedly, owing to the rather bulky construction of the engine and material used (aluminum 2024 with a thermal conductivity approximately 65 Btu/hr - ft^2 - $^{\circ}$F/ft) a large percentage of the heat released by the engine was dissipated to the mounting stand by thermal conduction.

The importance of exercising considerable care in shroud design is illustrated in Figure 15. When the air flow rates are identical, Shroud A shows an increase over Shroud B in cooling effectiveness ranging from 44 to 23 per cent, where effectiveness is defined as the actual air temperature rise of the air to be theoretical rise. The cylinder head temperature was used as the maximum temperature that the cooling air could attain. Figure 15 shows that the effectiveness for Shroud B is almost constant with cooling air requirements indicating that a large percentage of the cooling air by-passes the finned area of the engine. This is also evident in Figure 14 by the difference in heat removed by the cooling air for the two shrouds.

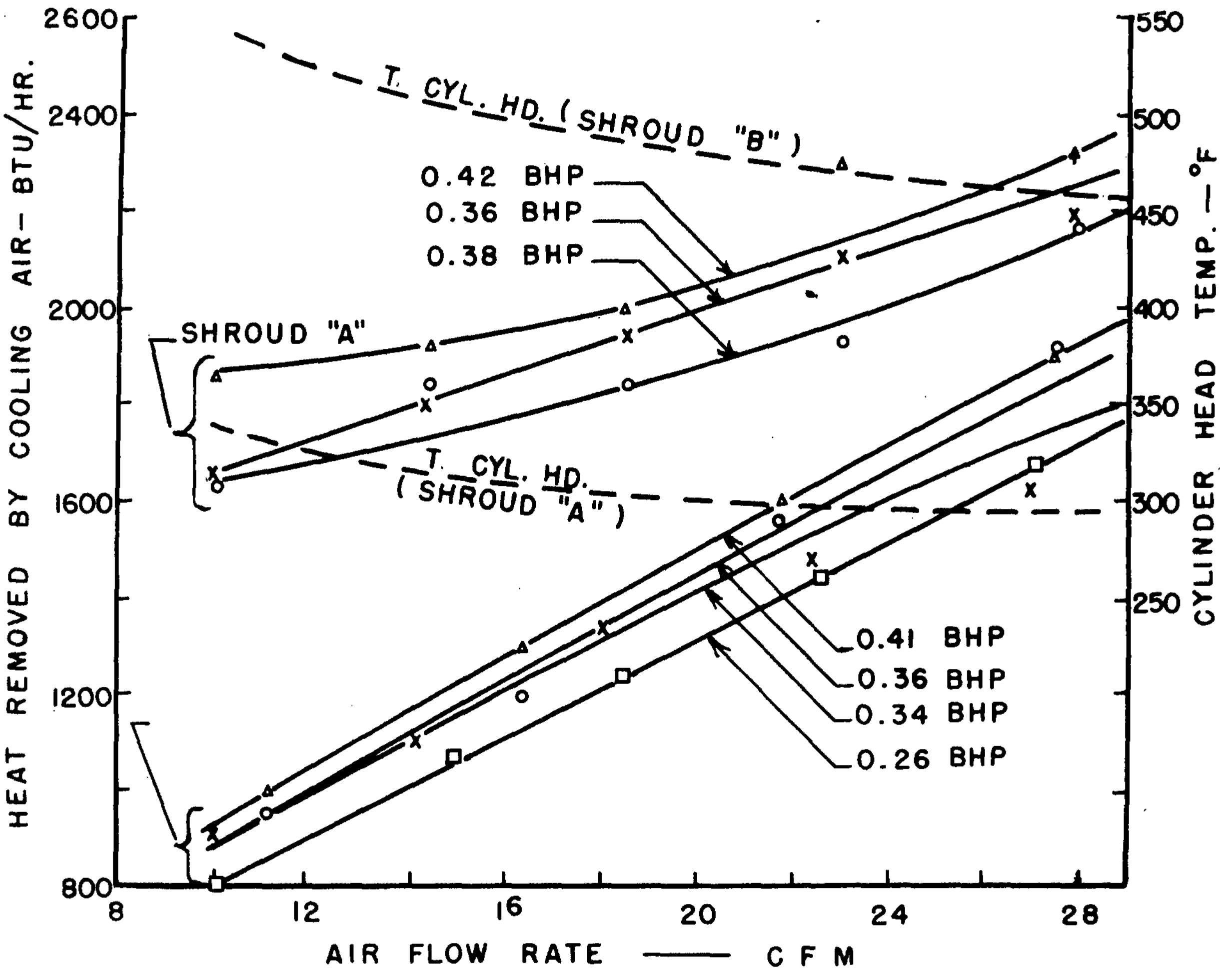


Figure 14. **Heat Dissipation to Cooling Air and Average Cylinder Head Temperatures for the Mark III, Two-Cycle, Cross-Scavenged Engine**

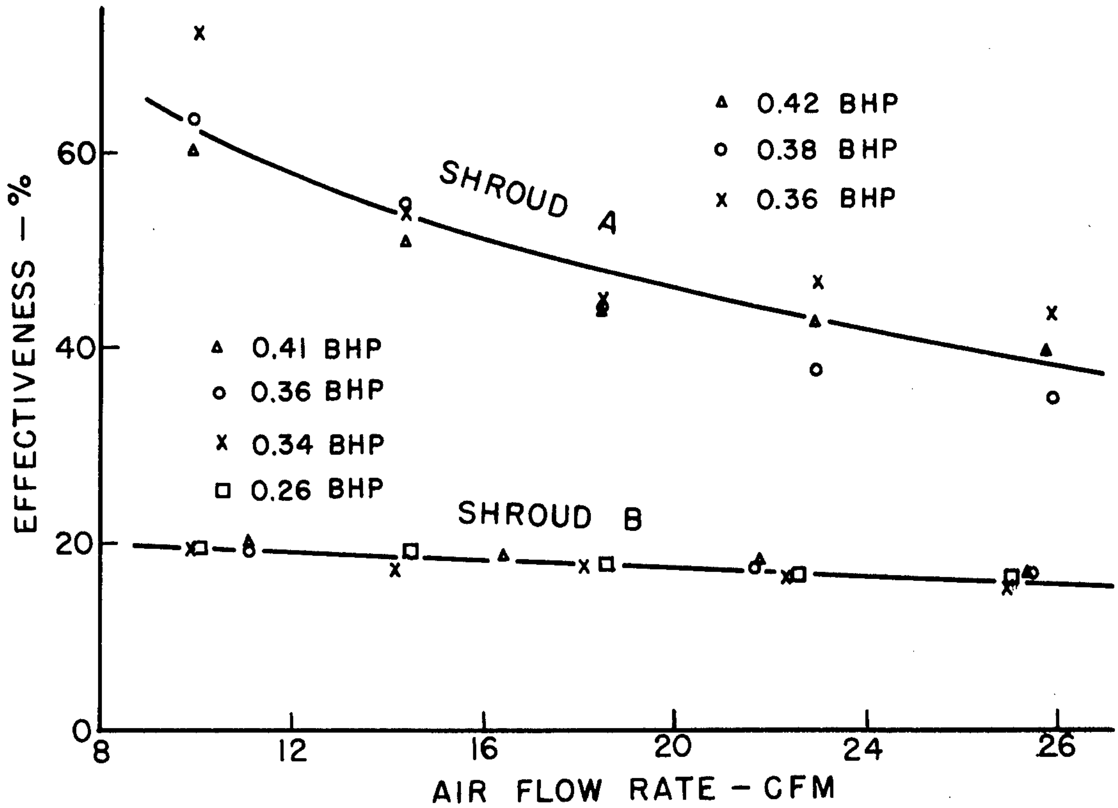


Figure 15. Effectiveness Curves for Mark III Engine With Two Different Shroud Designs

3.5.3 Four-Cycle L-Head

Figure 16 represents similar results for a four-cycle L-head test engine. These data are similar to the results for the two engines just discussed, except that there is more spread of the data with the four-cycle engine. As the curves show, the cooling load changed from 2,700 ± 400 Btu per hour at 17 cubic feet per minute to about 3,600 ± 400 Btu per hour at 28 cubic feet per minute. Thus, the average cooling load was about 3,100 ± 800 Btu per hour. The cooling load increased with engine load, except for the lowest engine output. With this engine there was a considerable increase in the cooling load with increase in the air flow rate.

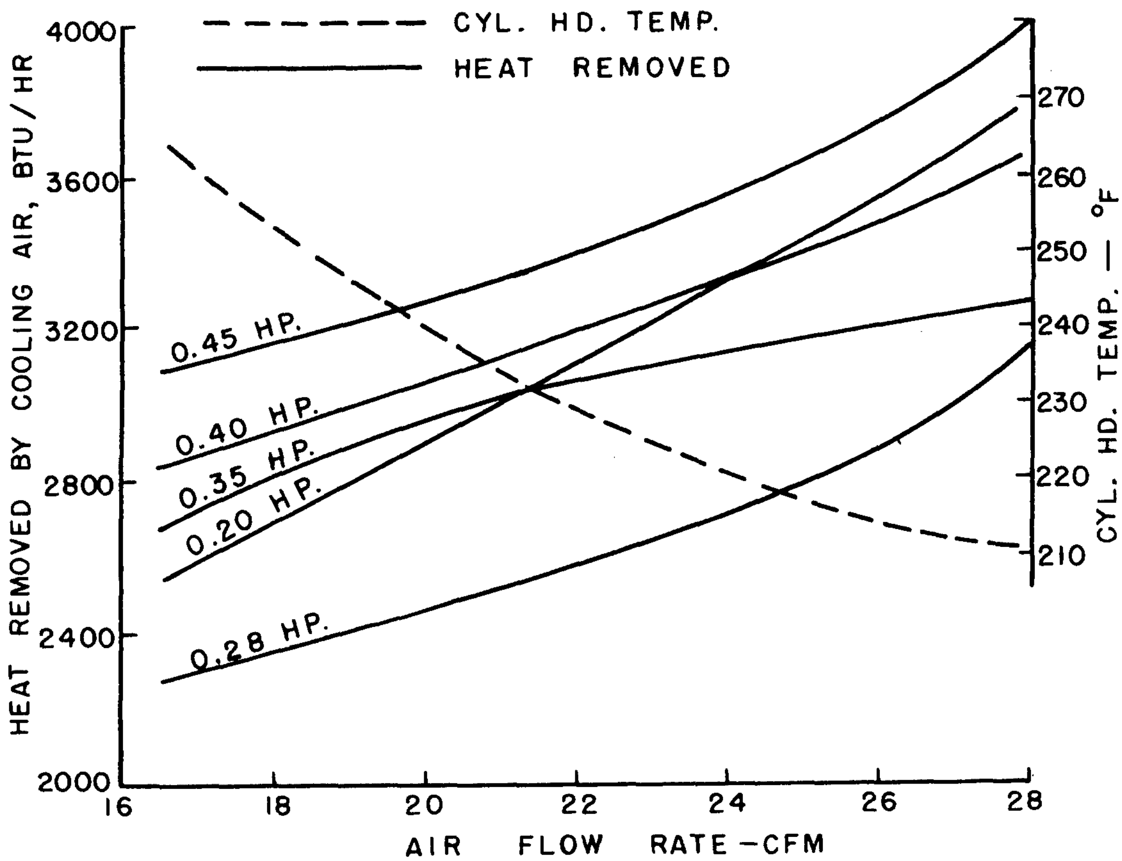


Figure 16. Heat Dissipation to Cooling Air and Cylinder Head Temperatures for a Four-Cycle, L-Head Engine

The difference in cooling effect in these three engines is probably accounted for in part by the great difference in cooling surface exposed to the air. The four-cycle, L-head engine was relatively bulky and had about 71 square inches of cooling surface. This effect is noticeable too in the cylinder-head temperatures, which are lower for the four-cycle engine. Because of the greater area of this particular four-cycle, L-head design, the cylinder-head temperatures could be kept much lower with the same air flow rates than with the other engines tested.

3.5.4 Liquid-Cooled Engine

The liquid-cooled, two-cycle test engine used in the test work was given the designation of the Mark II. Figure 17 and 18 are two photographs of this engine showing the basic construction. In Figure 18 the cylinder head has been removed showing the liquid passages. The complete cooling jacket was machined from 2024 aluminum with a total heat transfer surface of 25 square inches. Inlet and exit liquid temperatures were measured with iron constantan thermocouples located in the pipe tees seen in Figure 17. Inlet and exit liquid pressures were measured with calibrated pressure gages. Water was the coolant used in all test runs. The liquid was forced through the cooling jacket by a laboratory pump. The rate of coolant was controlled by varying the motor speed connected to the pump. Heat was removed from the liquid in a heat exchanger cooled by shop air.

The heat removed to the cooling liquid was of the same order of magnitude as for the Mark III, two-cycle test engine with the aluminum shroud. The rate of heat removal increased with liquid flow rate with an average value of 2100 ± 500 Btu per hour at a liquid flow rate of 100 pounds per hour. Similarly to the air-cooled engines the rate of heat removal of the Mark II increased with increasing engine load. Cylinder temperatures were more uniform with this engine than any of the other three tested. Head temperatures varied from 350°F (0.5 bhp) to 305°F (0.33 bhp) with a flow rate of 55 pounds per hour. With a flow rate of 290 pounds per hour, the corresponding cylinder head temperatures were 300 and 250°F.

The effectiveness curve shown in Figure 19 is an average value for four different series of runs made at various engine loads. Effectiveness is defined, in this case, as the ratio of liquid temperature rise to the temperature difference between cylinder head and inlet liquid. In comparison to the effectiveness attained with an air cooled engine, the average value is approximately 10 per cent higher than B and 15 per cent lower than Shroud A. Undoubtedly, the effectiveness could be appreciably increased by changing the design of the liquid flow passages.

In Figure 19 the superiority of liquid over air when considering only the cost of cooling is forcedly illustrated. The pumping power required to transfer the coolant through the cooling jacket is only a small fraction of that required to force air through a shroud. The ratio of pumping power to heat removed by the liquid increases rapidly as the flow rate is increased because of the increased turbulence in which results in larger pressure drops.

Figure 20 is a plot of the surface heat transfer coefficient plotted against pumping power in watts required to circulate the liquid or air for three of the test engines. The heat transfer coefficient is defined as the heat removed from the engine per unit of area divided by the temperature difference between cylinder head temperature and the average temperature of the cooling medium. As can be seen from the graph

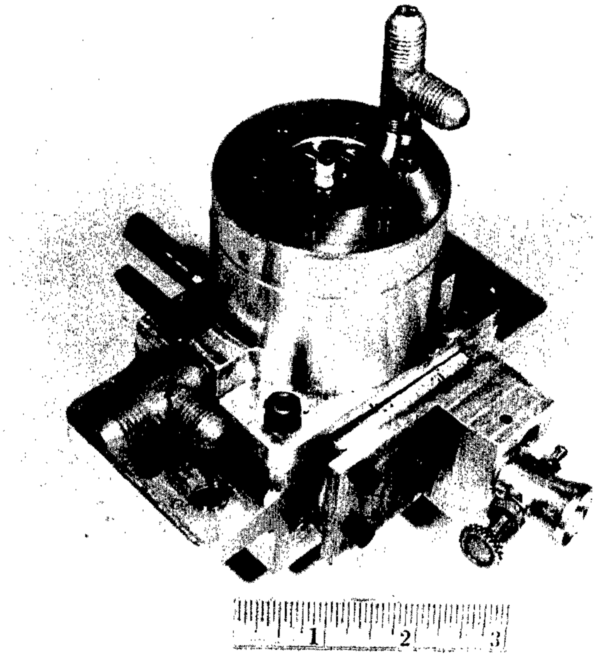


Figure 17. Assembled View of Liquid-Cooled, Two-Cycle Test Engine

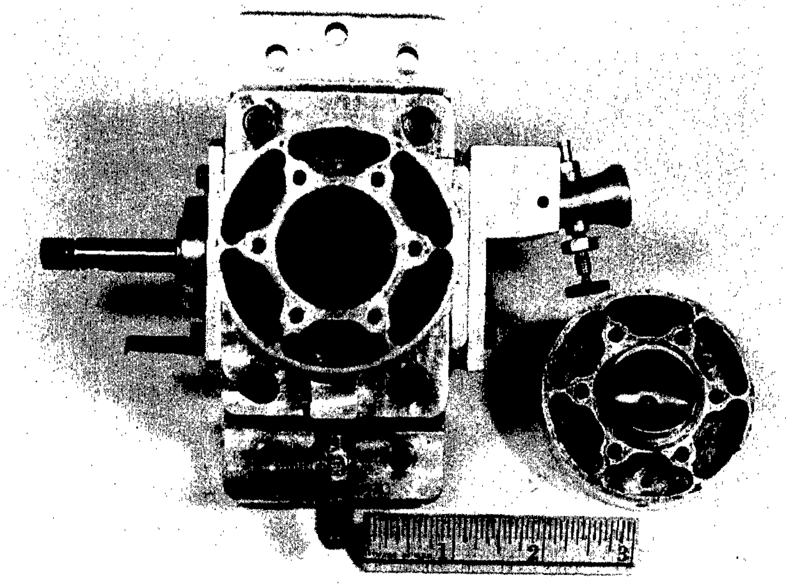


Figure 18. Head Removed Showing Cooling Liquid Flow Passages of the Mark II, Liquid-Cooled Engine

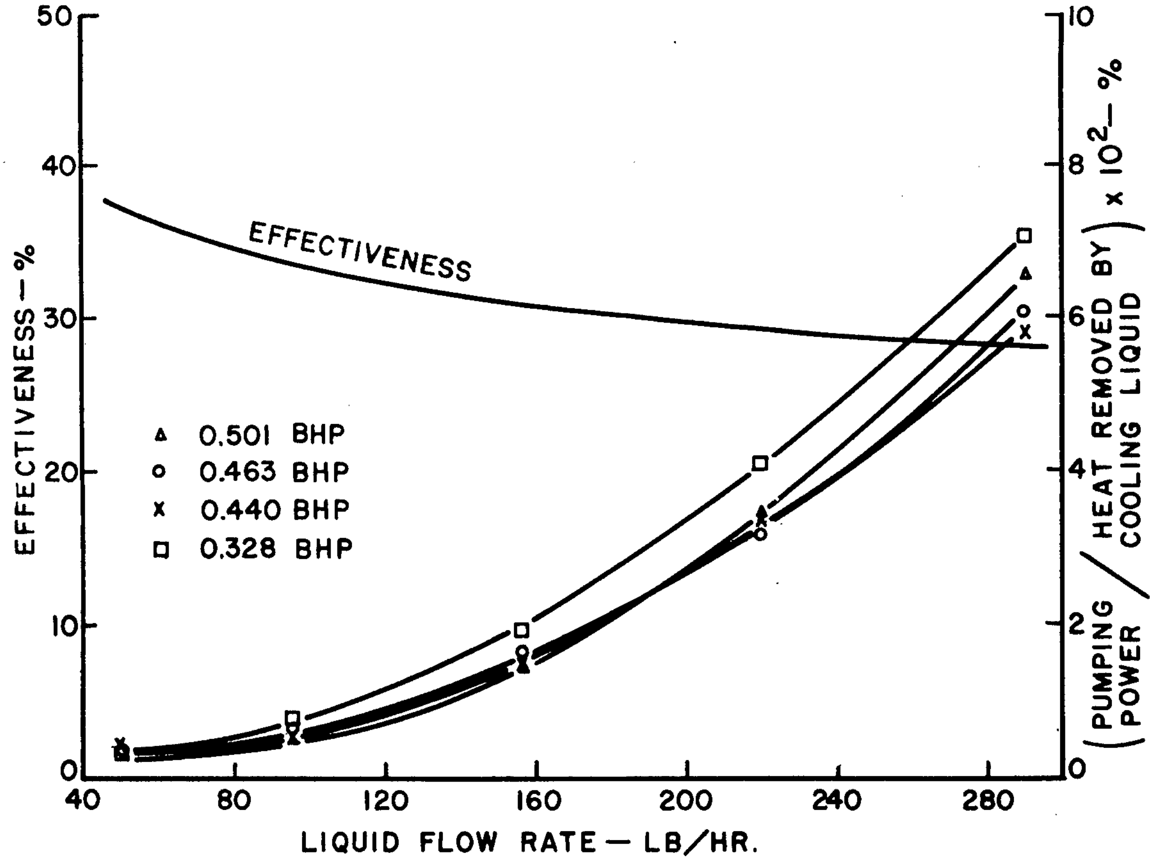


Figure 19. Effectiveness Curve and Ratio of Pumping Power to Total Heat Removed Curves for the Mark II, Liquid-Cooled Engine

the heat transfer coefficients obtained with forced liquid cooling are 10 to 40 times as great as those obtained by forced convective air cooling. However, when making an over-all comparison of liquid and air cooling, one should also consider the additional weight and complications of a liquid cooled system. Although thermally the liquid cooled system is excellent, the simplicity, reliability and compactness of an air cooled system with good shroud design appears to be superior for miniature engine-generator sets.

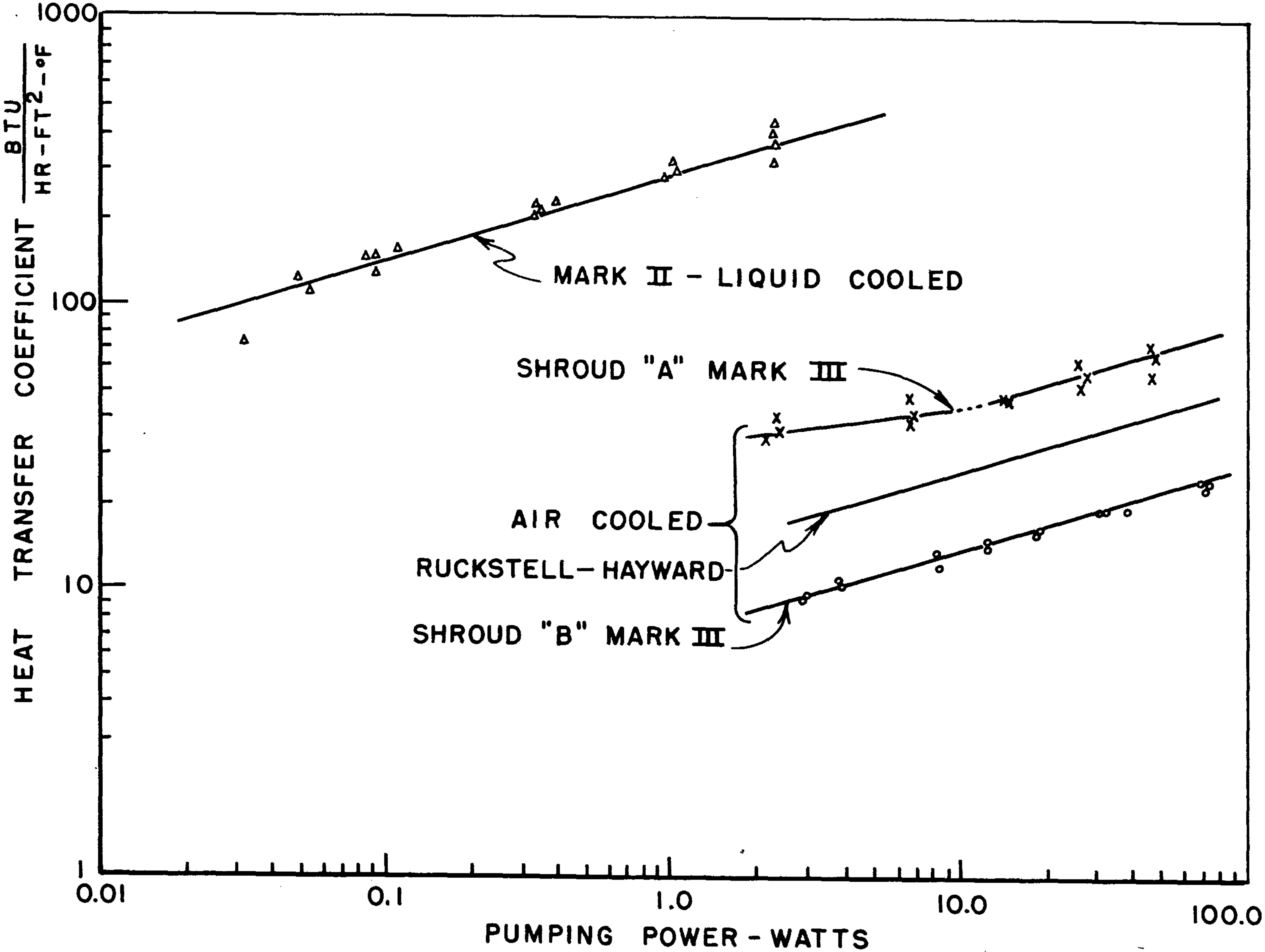


Figure 20. Surface Heat Transfer Coefficients for Two Air-Cooled Engines and the Two-Cycle Liquid-Cooled Engine

4.1 GENERAL

At the outset of the study of miniature engine-generator sets, the seriousness of the carburetion problem became evident. Evaporation or the lack of it, was considered to be a major limitation of the carburetors presently used on engines of the model aircraft size. Atomization, in the model engine carburetor, is produced by air blasting across a small orifice. The fuel flow is metered by varying the size of this orifice with a tapered needle. While this method has the advantage of simplicity and light weight, the orifice and flow passages are subject to fouling. Also, since the needle is not concentric with the orifice, the problem of adjustment is increased. The idling characteristics of a needle valve carburetor are poor.

Another carburetor studied during this project employed a constant area orifice of extremely small diameter to meter the fuel flow. Again the problem of fouling exists. A miniature fuel pressure regulator used in conjunction with a pressurized fuel tank to maintain a constant pressure at the carburetor regardless of the position of engine generator set proved to be unreliable because of fouling and diaphragm failure.

The apparent poor evaporation observed, mechanical troubles just described and high specific fuel consumption prompted the study of manifold design, hot spots, preheated air and/or fuel, atomization and mechanical design features of existing carburetors.

The following paragraphs are discussions of the results of various carburetion studies performed during the course of the project.

4.2 EVAPORATION RESULTS

4.2.1 General Remarks

Since the mechanical efficiency of the miniature engine is relatively high, it seemed as if the low thermal efficiency encountered may be due to poor utilization of the mixture. In the ideal case, the carburetor should meter the fuel in proper proportion to the air over the entire range of operating conditions. The mechanism for mixing the fuel and air, a nozzle in most cases, must provide fine atomization (minute droplets) promoting rapid evaporation.

The conventional carburetor used on larger engines performs satisfactorily over a wide range of operating conditions; but the major problem with smaller carburetors is to obtain reproducible flow characteristics, hence, reproducible evaporation characteristics, using small diameter jets.

From experience with larger engines it is known that as the liquid fuel is torn away from the jet, a portion of the fuel in droplet form is carried along in the air stream while the remainder forms a liquid film on the manifold walls. The liquid in droplet form and on the manifold walls evaporates until an equilibrium vapor pressure is reached. This type of evaporation is known as equilibrium air distillation, and the temperature of the resultant air-vapor mixture is termed the equilibrium air distillation (EAD) temperature. Figure 21 shows the relationship of EAD temperatures to the supplied air-fuel ratio and the air-vapor ratios formed at equilibrium. To reach the same percentage of evaporation at equilibrum, higher mixture temperatures are required for richer mixtures than for lean ones. At a given temperature, greater percentages of the leaner mixtures evaporate than for richer mixtures, but the richer air-fuel mixtures always yield the richer air-vapor mixtures. Figure 22 shows EAD temperatures for a variety of petroleum fuels (reference 2). Mixture temperature must be over 100°F for most fuels to attain 100% evaporation at one atmosphere. The temperature decreases as the manifold pressure decreases and increases for richer mixtures.

While the problem of evaporation in larger engines exists causing difficulties in cold starting, acceleration and mixture distribution it is amplified in miniature engines which use practically no manifold. Since the EAD curves represent the most favorable results which may be expected, the best possible design of manifold must be employed to hasten evaporation to equilibrium.

4.2.2 Factors Effecting Evaporation

The evaporation rate depends on numerous factors. Some of these are the temperature, viscosity, density, vapor pressure, and latent heat of the fuel; the temperature, pressure and velocity of the air; the air-fuel ratio; the rate of heat transfer to and from the manifold; the length and cross-sectional area of the manifold; and the degree of atomization obtained from the metering jet. The physical properties of the fuel by and large cannot be altered appreciably since a commercially available petroleum fuel of the proper volatility is desired. The size of engine and the operating conditions stipulate the air consumption, the manifold pressure, and the air-fuel ratio.

Thus, controllable factors of evaporation are limited to the temperature of the air, fuel, and manifold, and configurations of the mixing section and manifold. The results of studies which have been made of the effects of these factors on evaporation are as follows:

1. Heating the fuel until the liquid is capable of supplying the entire latent heat of vaporization leads to rapid evaporation, since there is no delay for such heat transfer. If the air is supplied at the EAD temperature, a typical petroleum fuel must be heated to about 400°F to give complete evaporation at the EAD temperature. Prohibitive fuel temperatures would be required to provide the latent heat if the air temperature were not at least equal to the EAD temperature (reference 2).

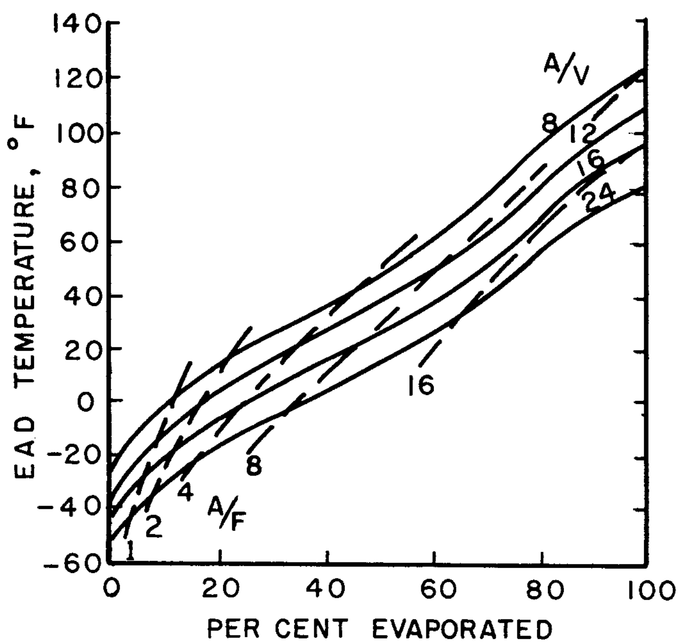


Figure 21. Relationship of EAD Temperatures to the Supplied Air-Fuel Ratio and the Air-Vapor Ratios Formed at Equilibrium

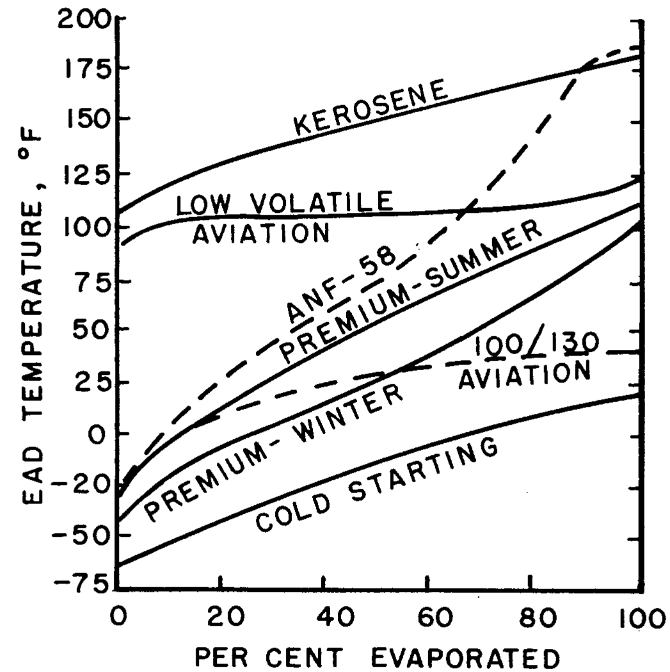


Figure 22. EAD Temperatures for Several Gasolines and Kerosene, 16:1 Air-Vapor Ratio, One Atmosphere Pressure

2. Air must be heated 40 to 50°F above the EAD temperature to provide the latent heat, if the fuel is supplied at the EAD temperature. However, since heat transfer from the air to the fuel is required, the temperature difference should be considerably greater than 50°F to obtain rapid evaporation.

3. Heat transfer from the walls of the manifold (or a hot spot) to the liquid or mixture is a function of the surface area presented, the coefficient of heat transfer, and the temperature difference between the surface and the mixture. The same factors, of course, govern transfer of heat from the air to the liquid. Rapid evaporation then depends upon a large surface area of liquid fuel, vigorous relative motion between the air and liquid, and an air or surface temperature considerably higher than the EAD temperature.

4. Use of a hot spot in the manifold is better than heating the inlet air to promote evaporation (references 4 and 26). Tests were conducted with an experimental manifold arranged so that the liquid drop would impinge against an electrically heated hot spot. For a motor gasoline, with mixture temperatures between 75 and 100°F, the hot spot evaporated about 5 per cent more fuel at a given mixture temperature than was achieved by heating the inlet air. Also, the hot spot gave complete vaporization at a lower mixture temperature. Heating the inlet air changed the air-fuel ratio metered by the carburetor slightly, and gave about 1 per cent lower volumetric efficiency than the hot spot heating.

5. Studies of the process of atomization of liquids in air streams, using photographic techniques, showed that a portion of the liquid mass was drawn out into a fine ligament, which was quickly cut off by the rapid growth of a dent in its surface. The small detached mass then swiftly formed a spherical drop. At higher air velocities, the ligaments became smaller; they existed for shorter time intervals, and they formed smaller drops. Also, at higher air velocities there was less difference in size between the smallest and largest drops. With water the drop size decreased with increased air velocity up to about 330 to 400 ft/sec. Higher velocities gave no appreciable further reduction in drop size. Alcohol, with a lower surface tension than water, reached a minimum drop size at lower air velocities (reference 8).

6. Other studies, using the relative transparency of the air-fuel mixture to determine the degree of atomization, showed that (a) with increased manifold vacuum, the mean drop size and the quantity of fuel in the liquid film on the wall both decrease, (b) the shape of the fuel nozzle had little effect on atomization, and (c) small changes in the diameter of the venturi did not affect atomization (reference 36).

7. A mathematical analysis of evaporation of small drops indicated that the rate of evaporation of a drop in a gas atmosphere, as opposed to evaporation in a vacuum, is proportional to the diameter and not the surface area of the drop, and that the evaporation rate is such that the drop's surface area changes linearly with time. This relationship is

true, down to a certain drop size, for widely different experimental conditions. Very small drops evaporate in a gas atmosphere with the same rate as in a vacuum (reference 11).

8. Work of a more quantitative nature showed that the degree of atomization may be characterized by (reference 37):

$$K = \frac{\rho U^2}{T}$$

where

K = atomizing characteristic, cm^{-1}
ρ = density of liquid, g/cc
U = velocity of air relative to the liquid, cm/sec
T = surface tension, dynes/cm.

The fineness of atomization is proportional to the value of the characteristic, K. Alcohol (which has a surface tension about 1/3 that of water) showed significantly better atomization than water at the same air velocities, but similar atomization at similar values of K.

9. Atomization is very poor with water when values of K are less than 120, corresponding to air velocities of about 90 ft/sec (reference 37). Photographs indicated heavy precipitation on the walls with K = 120. Observations with a comparator showed that 90 per cent of the liquid was atomized into drops 0.18 mm diameter or less when the velocity was 5330 cm/sec (175 ft/sec), K = 1420/cm; while at 1550 cm/sec (51 ft/sec), K = 120/cm, 90 per cent of the fuel was in drops of 0.8 mm or smaller. In general, to give flexible engine operation, K should be 400 or higher, which corresponds to air velocities in excess of 100 ft/sec.

10. The shape of the venturi is relatively unimportant; more is to be expected from improvements in the fuel nozzle design (reference 37).

11. Nukiyama and Tanasawa present the following empirical expression for the mean drop size formed by atomization (reference 25, 28).

$$d_o = \frac{585\sqrt{T}}{U\sqrt{\rho}} + 597\left(\frac{\mu}{\sqrt{T\rho}}\right)^{0.45}\left(\frac{1000\,L}{A}\right)^{1.5}$$

where

d_o = diameter of single drop with same ratio of surface to volume as total sum of drops, microns
U = velocity of air relative to liquid, meters/sec
T = surface tension, dynes/cm
ρ = liquid density, g/cc
μ = liquid viscosity, poises
L/A = ratio of volume of liquid to volume of air at throat.

This equation is not dimensionally consistent but it holds reasonably well for atomization in a venturi for values of $T = 19$ to 73 dynes/cm, $\rho = 0.7$ to 1.2 t/cc, $\mu = 0.003$ to 0.5 poises, and U less than sonic velocity. Other investigators have shown that the equation agrees fairly well, and to the same order of magnitude at least, for atomization in other types of atomizing devices (reference 21). It will be noted that when the ratio of air to liquid is large, as it is in a carburetor, the atomization is governed mainly by the first term in the equation and the viscosity is of minor importance. The square of the reciprocal of this first term is proportional to the atomization characteristic K discussed above in paragraph 8, which verifies that evaporation is a function of the surface area of the mean drop.

12. Experiments with high-velocity vaporizers using high-boiling-point liquids indicated that evaporation can be obtained in 0.002 to 0.07 sec, depending on the gas velocity and the initial drop size. A venturi tube was used with gas velocities from 500 to 1800 ft/sec to atomize 25° API oils having an average molecular weight of 380. Log mean temperature differences as high as 600°F between the air and oil were used successfully (reference 9).

13. Centrifugal sprays may yield more uniform distribution of drop sizes than a plain spray nozzle (reference 20). This and the preceding paragraph suggest that some innovations foreign to usual practice in the engine field might be the answer to rapid evaporation in miniature engines.

14. Little quantitative work has been done concerning the evaporation of liquid drops in turbulent air streams. However, the same parameters which govern heat transfer apply to the evaporation process, so that rapid evaporation would be obtained with high relative velocities, high log mean temperature differences, and good atomization to expose a large liquid surface. Evaporation of the drops is actually relatively unimportant, because satisfactory engine operation probably is achieved whether the atomized drops evaporate or not. Evaporation of the liquid film deposited over the inlet system, however, seems to be a principal key to good engine performance.

Evaporation rates for liquids from plane surfaces, and inside pipes with flow parallel to the surface, are expressed by the following equation (references 22, 28), for Re above 20,000:

$$\frac{k}{G}\left(\frac{\mu}{\rho D}\right)^{2/3} = \frac{0.036}{(Re)^{0.2}} = \frac{0.036}{\left(\frac{xG}{\mu}\right)^{0.2}}$$

where

k = mass transfer coefficient, lb/(hr)(sq ft)(mole fraction)
G = mass velocity, lb/(hr)(sq ft)
μ = viscosity of liquid, lb/(ft)(hr)
ρ = density of liquid, lb/cu ft

D = diffusivity, sq ft/hr

Re = Reynold's Number, in which the length term is measured in the direction of flow

x = total length of flow passage, approach length plus wetted length, ft (for a carburetor, this length could probably be measured from the throat to the end of the wetted surface).

Rearranging,

$$k = \frac{0.036\ G^{0.8}\ (\rho D)^{0.67}}{x^{0.2}\ \mu^{0.47}}$$

The rate of evaporation from the liquid film would be a function of the mass transfer coefficient, the wetted area, and the concentration of fuel vapor in the manifold:

$$\text{evaporation, lb/hr} = k(\pi dx)(\text{mole fraction of air}),$$

where

$$d = \text{diameter of manifold, ft.}$$

The mole fraction of air in the typical air-vapor mixture supplied to most engines is about 0.98 and would vary only slightly with operating conditions. Likewise, the viscosity, density, and diffusivity of the liquid are dependent upon the hydrocarbon fuel selected, and can not be altered conveniently except with changes of fuel temperature. Dropping these terms from the equations,

$$\text{evaporation, lb/hr} \approx dG^{0.8}\ x^{0.8}.$$

Substituting $W/(0.785\ d^2) = G$, where W = mass flow rate of air, lb/hr,

$$\text{evaporation, lb/hr} \approx \frac{(Wx)^{0.8}}{d^{0.6}}\ .$$

Thus, for an engine having a fixed air consumption, good evaporation would be obtained in a manifold having a large ratio of length to diameter. Almost the opposite condition prevails in most small two-cycle engines, wherein the mixture is admitted to a crankcase of relatively large effective diameter and a short length of flow passage.

4.2.3 Evaporation Experiments

An evaporation experiment was performed using the apparatus shown in Figure 23. The technique for measuring the amount of evaporation by analyzing the combustion products of a burned sample of the air-vapor mixture proved unreliable. An alternate method, that of observing the relative amounts of liquid film on the walls of the glass manifold,

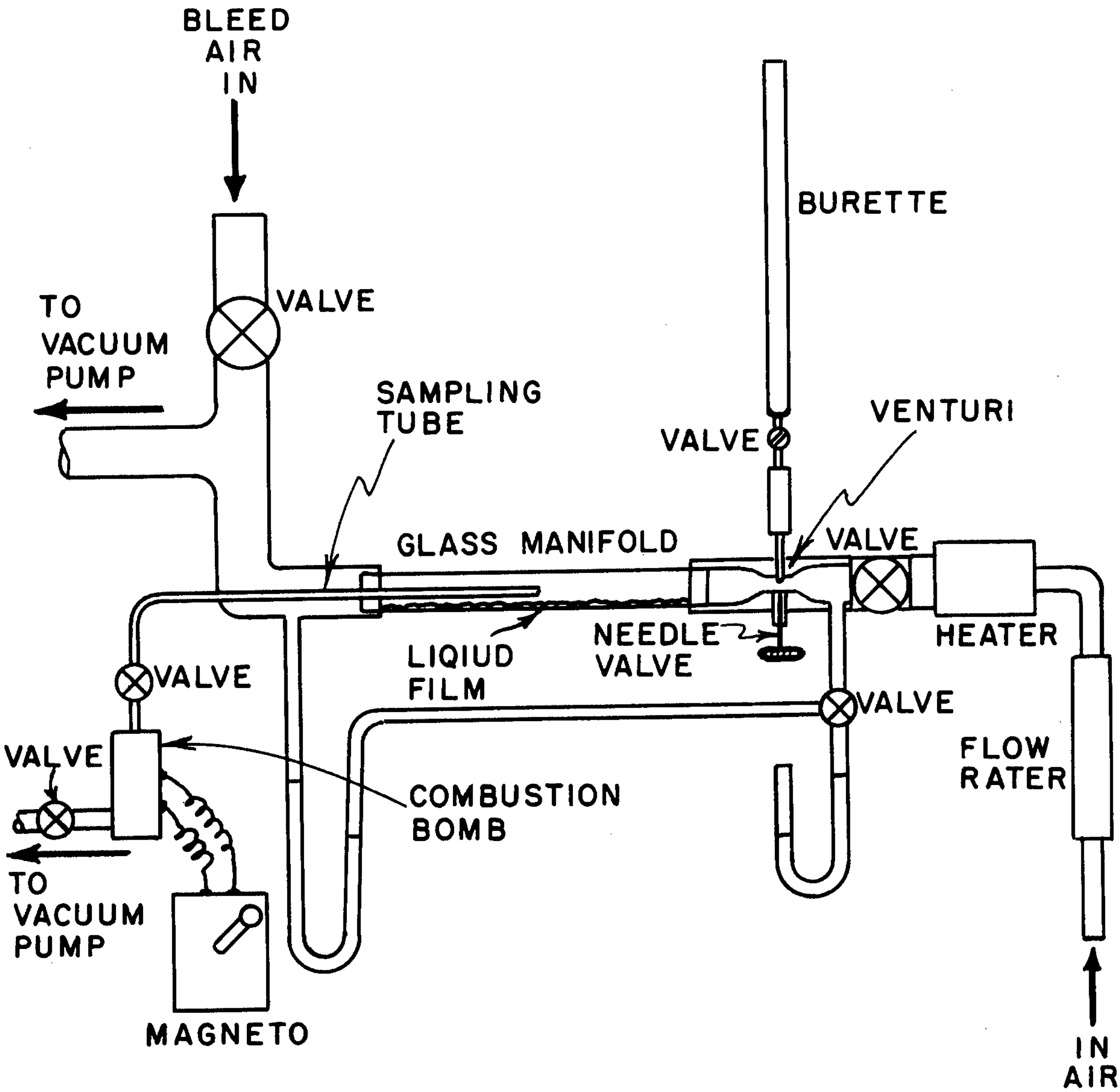


Figure 23. Apparatus Used for Evaporation Studies

provided some preliminary information relative to evaporation. This information appears in Table C. The experiment was conducted using regular motor gasoline, with glass manifolds about 15 in. long, and a venturi diameter of 0.406 inch.

TABLE C. RELATIVE DEGREES OF EVAPORATION FOR DIFFERENT MANIFOLD CONDITIONS

Inlet Air			Manifold						
Pressure In.Hg Abs	Temp °F	Flow lb/hr	Velocity ft/sec	Evap *	Pressure In.Hg Abs	Temp °F	Flow lb/hr	Velocity ft/sec	Evap *
For 0.395 in. I.D. Manifold					For 0.485 in. I.D. Manifold				
28.7	89	17.0	80.6	3	28.8	88	8.5	26.8	2
28.7	123	17.0	85.7	3	28.8	163	8.5	30.3	2
28.7	150	16.8	88.5	3	28.8	182	8.6	31.7	3
28.7	185	17.3	96.5	4	23.6	88	8.5	32.8	2
23.5	89	17.0	98.9	3	23.9	161	8.5	36.5	2
23.6	122	17.0	104.7	3	23.8	182	8.5	38.1	3
23.4	150	16.8	110.4	4	18.7	88	8.4	40.7	2
23.5	180	16.9	115.2	4	19.0	159	8.6	46.5	2
18.6	89	16.9	125.2	3	18.4	186	8.3	48.2	3
18.6	122	16.9	133.3	4	13.8	88	8.5	55.9	2
18.7	151	17.0	140.0	4	14.0	133	8.5	59.6	2
18.4	179	16.9	149.8	4	13.4	160	8.5	65.1	3
13.4	89	17.1	178.9	4	13.8	194	8.1	63.6	3
13.6	120	17.0	185.0	4	28.8	91	17.0	53.7	2
13.7	152	17.1	196.4	5	28.8	157	16.7	59.2	2
13.7	178	17.2	198.9	5	28.8	185	17.0	63.0	3
28.6	93	33.8	164.5	4	23.9	91	16.8	63.8	2
28.6	114	33.4	169.6	4	23.8	152	16.9	71.7	3
28.6	148	34.1	183.0	4	23.9	181	17.1	76.2	3
28.6	180	33.8	192.0	4	18.8	91	17.0	82.4	2
23.9	92	34.0	200.4	4	18.2	127	17.0	91.0	3
23.9	115	34.1	210.3	4	18.8	181	16.8	95.2	3
23.9	151	34.1	223.7	4	13.7	91	17.1	113.8	3
23.5	180	33.9	238.0	4	13.7	154	17.2	128.1	3
18.9	92	33.8	259.0	4	14.1	180	17.0	128.4	4
19.1	118	33.8	268.0	4	For 0.859 in. I.D. Manifold				
19.7	151	33.8	275.3	5	28.9	184	8.4	9.9	1
19.7	179	33.7	287.3	5	21.8	183	8.5	13.2	2
28.4	92	42.4	209.7	4	14.4	92	8.5	17.2	1
28.4	117	42.4	219.5	4	14.4	183	8.6	20.2	2
28.3	143	42.5	235.0	5	28.8	185	17.5	20.4	2
24.1	92	42.5	256.8	4	21.8	88	17.2	22.6	1
24.1	120	42.7	271.0	4	22.1	182	17.2	26.4	2
24.1	139	42.5	277.7	5	14.2	152	17.0	38.8	2
24.5	178	42.5	291.7	5	14.3	181	17.1	40.3	3

*Relative degrees of evaporation, coded as follows:

1. Entire manifold covered with liquid film, heavy flow on bottom
2. About 2/3 of manifold covered with film,fair flow on bottom
3. About 1/3 of manifold covered with film, some flow on bottom
4. Little wall film, little flow on bottom, few drops in suspension
5. No wall film, only few drops in suspension at end of manifold.

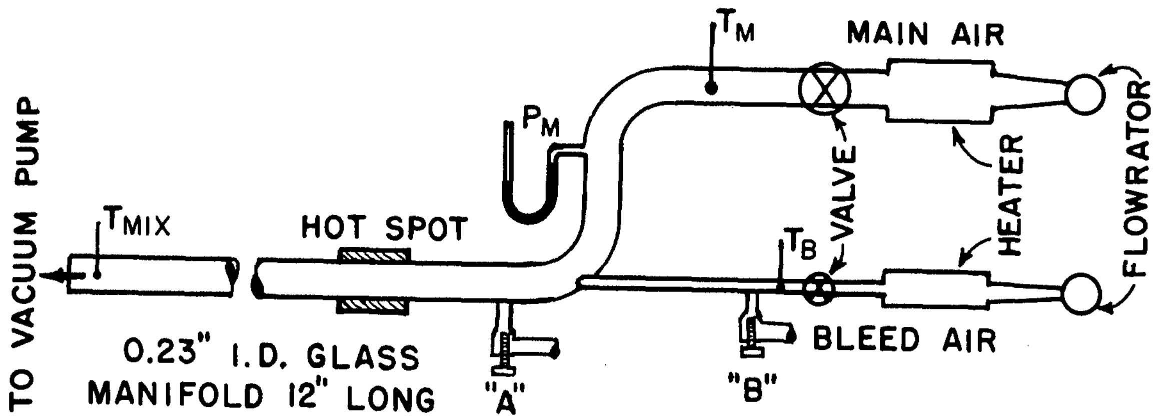


Figure 24. Experimental Carburetor

An experimental carburetor, utilizing the manifold configuration employed in the above evaporation experiment, was designed and constructed. The apparatus is shown diagramatically in Figure 24. All tests were made with a vacuum of about 0.5 in.Hg in the manifold, with air-fuel ratio 13.5 to 1 by weight, and with a total air flow rate of 2.0 cfm. These conditions gave manifold velocities of the order of 125 to 150 ft/sec, which were conducive to good evaporation if reasonable atomization was obtained in the carburetor. Fuel was admitted either at the main venturi (A) or at the bleed venturi (B). The flow rates, temperature, and pressure of the main air and the bleed air could be controlled independently of each other. The relative amount of evaporation achieved with the different test conditions was determined by visual inspection of the mixture in the glass manifold.

Preliminary tests indicated that mixing the fuel with bleed-air at venturi (B) promoted rapid evaporation, even with relatively low mixture temperatures leaving the glass manifold. Varying the percentage of bleed air used between 12 and 20 per cent did not affect evaporation, but best results were achieved when the bleed air was heated to 200° to 400°F. When the fuel was admitted at venturi (A), good evaporation could be attained

only by heating the main air to a high temperature, which resulted in mixture temperatures higher than would be desirable for good volumetric efficiency. The amount of bleed air used had a noticable effect on evaporation, probably due to the influence of the bleed air on the resulting temperature of the mixture. Comparable evaporation was attained when the fuel was admitted at (A) and no bleed air was used, but in no case, for equal mixture temperatures, was the evaporation as good as when the fuel was admitted at (B). The few tests conducted seemed to show that it is advantageous to mix the fuel with high temperature bleed air well ahead of the main venturi, as at (B).

Later tests with an electrically heated hot spot indicated the hot spot aided evaporation.

Actual engine testing seemed to indicate the problem of evaporation was not as serious as it was originally thought to be. A miniature engine was tested using a pre-mixed air-vapor mixture as opposed to using the standard needle-valve carburetor. Preliminary tests have shown essentially no differences in fuel consumption between the two systems. Power tests were conducted on both four-cycle and two-cycle engines using fuels having boiling points ranging from -44°F to 210°F. The optimum brake specific fuel consumption observed in these tests ranged from 0.6 to 0.75 lb/bhp-hour. Apparently good evaporation had been achieved with both fuels since the highly volatile fuel (boiling point -44°F) did not produce an appreciably lower BSFC or higher power output. Fuels of lower volatility, such as kerosene and jet engine fuels, produced a BSFC slightly higher than that observed for gasoline or propane.

On the basis of these tests it was concluded that the loss in power and fuel due to insufficient evaporation of gasoline was negligible; small engines, two or four cycle, of simple design using the standard needle valve carburetor for aviation gasolines will evaporate the fuel satisfactorily and produce good power and BSFC; hot spots, turbulence, long manifolds, high air and fuel temperatures and premixing air with fuel (air-bleed) promote good evaporation but are of little value except to ease starting at normal and low temperatures.

4.3 CONTROL OF FUEL FLOW RATE

4.3.1 General

An important function of any carburetor is uniformly metering the fuel in the proper ratio to the air flowing through the engine at various operating conditions. The metering function of a miniature engine carburetor is extremely critical. So that some idea of the problem involved in metering small quantities of fuel the following example is presented.

A miniature engine is to develop 0.25 bhp at 16,000 rpm with a BSFC of 1.5 lb/bhp-hr. The fuel flow rate at these conditions is 0.375 lbs/hr or 0.0001 lbs/sec, which means in order for the engine to burn one pound

of fuel it must complete 2,560,000 cycles. The amount of fuel burned each cycle is almost the same; hence a pound of fuel must be divided into 2,560,000 equal parts. The example indicates that not only must the fuel metering element produce a uniform flow, it must also be sensitive enough to correct slight variations in the flow.

Since slight variations in air-fuel ratio will cause misfiring and eventually render the engine inoperative, uniformity of fuel flow and fast response to changes in air demand are essential for optimum performance and reliability.

4.3.2 Commercial Carburetors

The modern carburetor, consisting of a float bowl, venturi, main metering jet, compensating jets for idling and acceleration, has been developed to the state where it will give satisfactory performance, providing the engine tilt is kept to a minimum. Although there seems to be no great difficulty in adapting a conventional carburetor to the miniature size, it is the opinion of carburetor manufacturers that the task of matching a carburetor of new design to a given engine will require a rather lengthy development program. Matching a carburetor to an engine-generator set of low speed and a power output of 400 watts or more would not be as difficult as would be in the case of a high speed, 50 watt engine-generator. Minute flow passages, small orifices, small light weight pressure regulators, precise machining, etc. are results of miniaturization and may cause some difficulty in acquiring a reliable product. The problem is further complicated by such requirements as the ability of the engine to be started and run in any position, automatic starting and at environmental extremes of temperature, moisture, etc.

Considerable time was spent on this project in studying devices that might best fulfill the requirements of engines that are not always level. Carburetion systems utilizing variable area constrictions, variable velocity, variable pressure and air-bleed as methods of controlling the fuel flow are discussed in the following section.

4.3.3 Experimental Designs

From the study of construction features of many conventional carburetors it seemed that the most desirable means of metering fuel was by the use of a venturi. The venturi in the air passage of a conventional carburetor is the sensing element for changes in air flow to the engine. An increased air flow due to an increase in engine speed will produce a decrease in pressure at the throat of the venturi. Located at the venturi throat is the main fuel metering jet or nozzle. As the throat pressure decreases, the pressure drop across the fuel jet increases, thus increasing the fuel flow. In large engine carburetors the fuel flow is altered somewhat by the use of stepped metering rods, a device used to change the area of the main jet orifice. The purpose of all this is to secure the proper air-fuel ratio required by the engine under conditions of idling, economy range (medium power), and full power.

Fuel injection either into the manifold or the combustion chamber was considered briefly as a means of metering the fuel. However, because of the complexity and inherent problems associated with miniaturization, it was felt that such a device would require considerable time in development to achieve the desired degree of reliability.

A device called a spiral flow regulator, Figure 25,26, was designed and built for initial tests on fuel metering. It utilizes a flow passage of variable flow resistance to obtain the desired metering. The fuel is supplied to the metering section at a constant pressure and the flow rate is controlled entirely through the variation of the flow resistance through the metering section. The flow resistance is controlled by adjusting the length of spiral through which the fuel must flow before it is admitted to the inlet air stream.

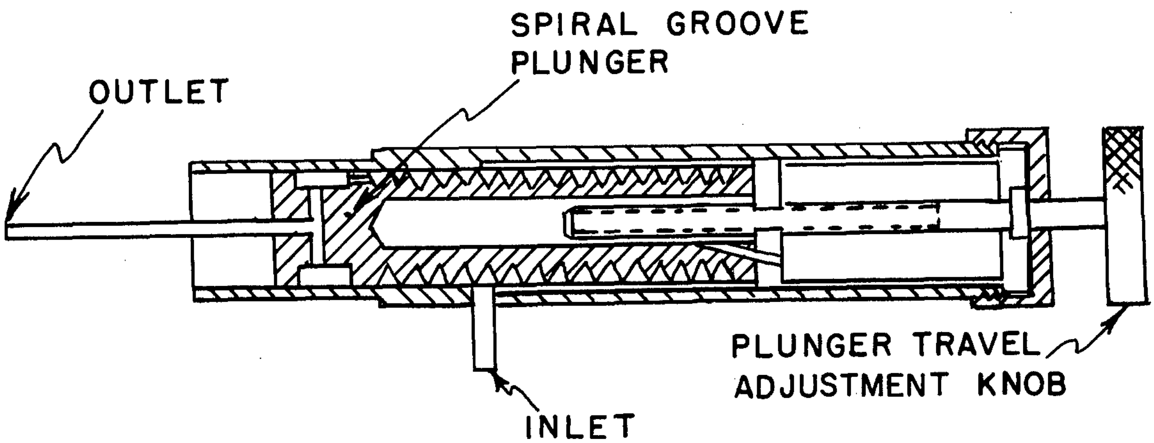


Figure 25. Sectional View of Experimental Fuel Regulator

Since this experimental model was not used in conjunction with a venturi, a manual adjustment of the plunger was required to vary the flow rate. With considerable refinement this design appears to be well suited forgovernor control. Either a mechanical governor with suitable linkages or an air flow sensing device (venturi and diaphragm system) might be used to move the plunger. The effect of small particles of dirt in the fuel system would not be nearly as disastrous as in present miniature carburetors, since the metering passage is larger, but very long, and the governor would adjust the plunger to admit more fuel if the passage became slightly plugged.

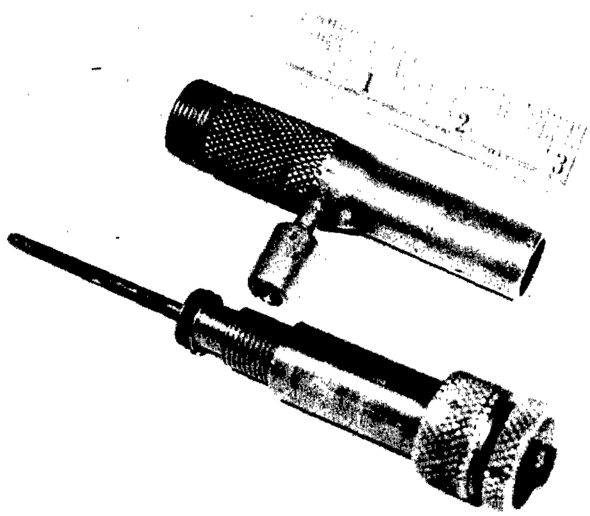


Figure 26. Partial Disassembly of Experimental Fuel Showing Body, Spiral Groove Plunger and Fuel Inlet and Outlet Passages

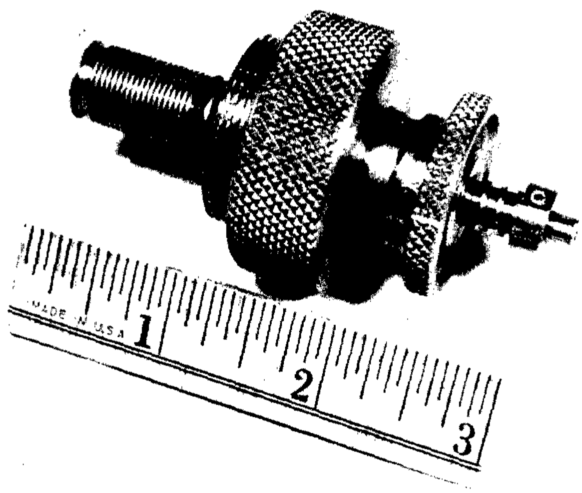


Figure 27. Redesigned Spiral Flow Regulator Integral With a Pressurized Fuel Tank (not shown)

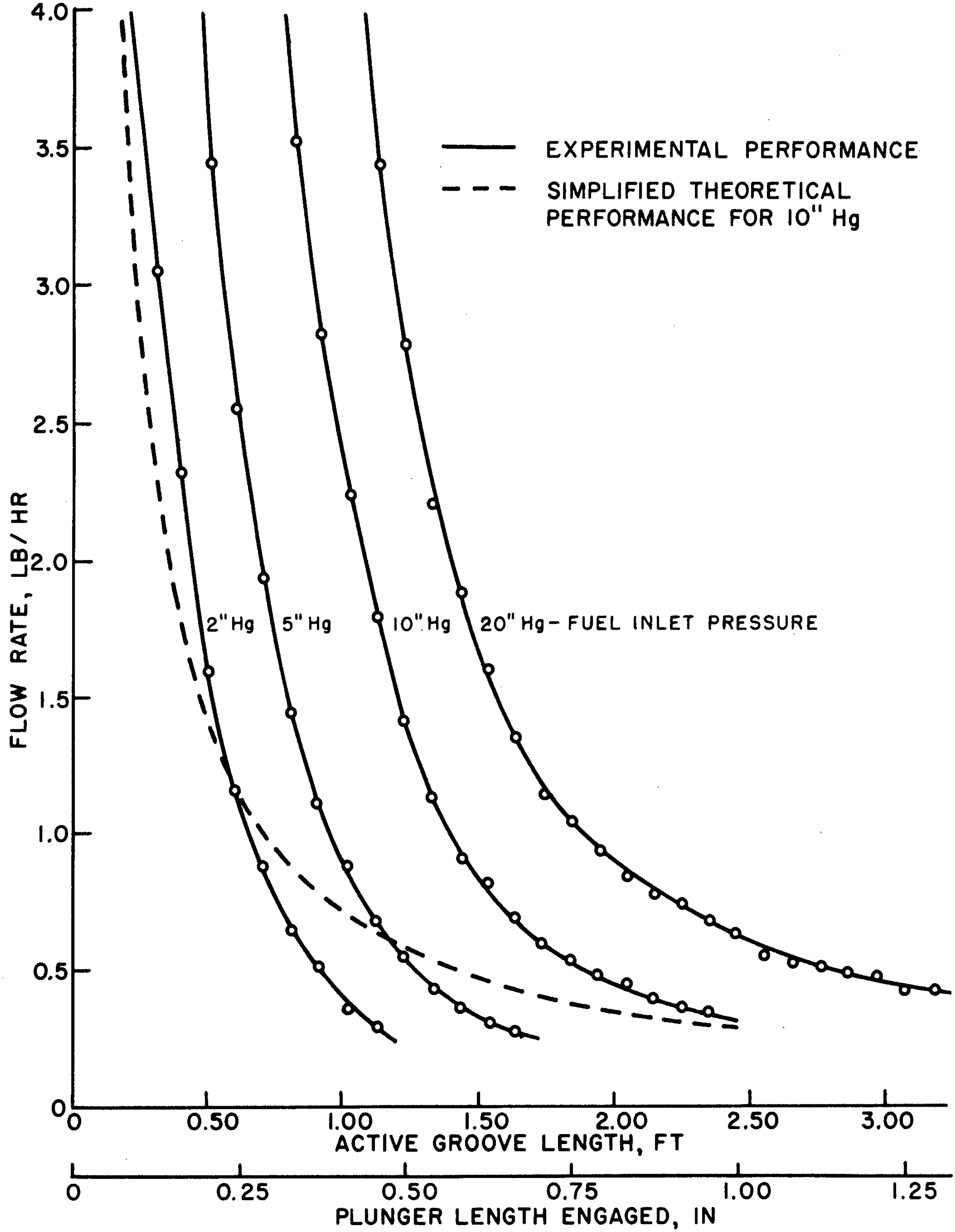


Figure 28. Performance of Experimental Flow Regulator at Various Inlet Pressures Using Water as the Fluid

Figure 28 shows the metering characteristics with water at several different inlet pressures. The great difference between the theoretical and observed flow rates at the higher flow rates and inlet pressures, is due to leakage across the lands. However, this difference is less at lower flow rates and inlet pressures because of the labyrinth effect attained between the lands and cylinder walls.

Satisfactory engine operation was achieved with this method of fuel regulation but only for a short time after starting. Air leaked into the system forming large bubbles in the fuel line which eventually caused the engine to stop.

Redesign of the regulation system incorporated the spiral groove plunger integral with a pressurized fuel tank (Figure 27). The plunger does not come in contact with air, thus eliminating the possibility of air entering the line. The flow characteristics of the new model are similar to the original and the relative ease with which the fuel flow can be controlled remained the same.

In both experimental designs the friction due to the "O" ring plunger seals was rather high. Since only small amounts of power are available to move the plunger and associated linkages in a governing system, low friction is essential. It is conceivable, however, that a device utilizing the flow resistance method could be designed with friction kept at a minimum.

Further consideration of the fuel metering problem lead to a carburetor design utilizing the principle of variable pressure as the means of controlling fuel flow shown in Figures 29, 30, 31. The fuel nozzle, which is concentric with the convergent section of the venturi, employs a needle valve to initially adjust the fuel flow. Once this adjustment has been made, the flow rate is regulated by varying the position of the nozzle. This is possible since, at a constant engine speed, the mass flow of air is constant and the static pressure in the convergent section decreases from atmospheric to a minimum at the throat. This method of controlling the fuel flow was found to give easier control of engine speed. The idling characteristics of the carburetor are an improvement over those of the standard needle-valve, air-horn type carburetor. Power output and BSFC of an engine using the experimental carburetor were comparable to the same engine using the standard needle-valve carburetor.

The possibility of automatically changing the position of the fuel nozzle, thus changing fuel flow to meet the air-fuel requirements of the engine at various loads, seemed promising. Since the friction was very low, an air pressure sensing diaphragm or mechanical speed governor with appropriate linkages could be used to position the nozzle. A diaphragm, sensing the air pressure changes in the venturi throat, was built and tested. Although the initial results were unsatisfactory, it seemed that further development could produce a reliable system.

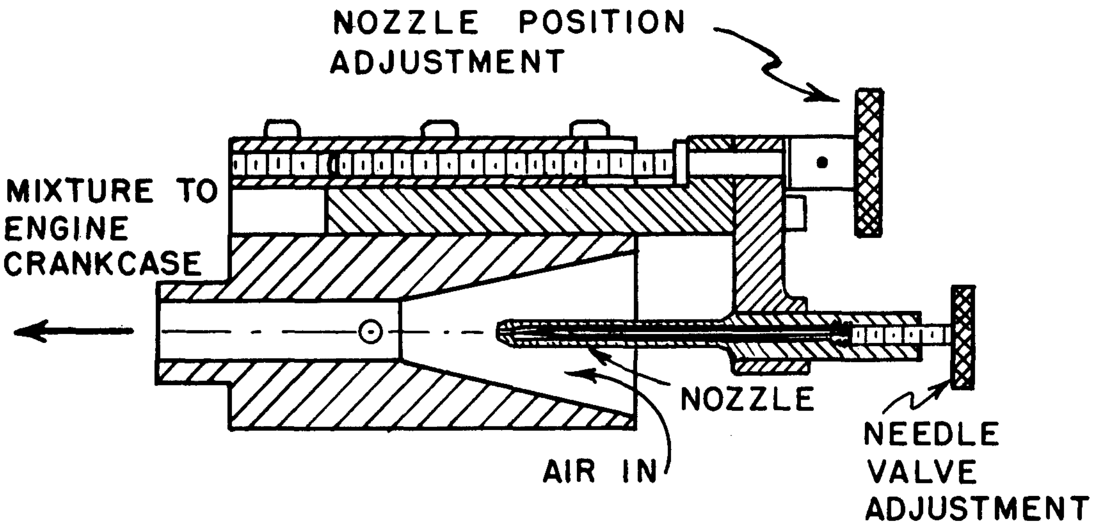


Figure 29. Sectional View of Experimental Variable Pressure Carburetor

It was suggested the additional modifications might include an air-bleed mechanism. An air inlet hole could be drilled in the nozzle and, in place of the longer needle valve now used to initially regulate the fuel flow, a shorter one would be used to regulate the air flow. Better control of the fuel flow may be realized.

Because of the many outstanding features of propane as a fuel for miniature engines considerable time and effort were expended in designing and developing a reliable propane carburetor. It was designed to give good control of air-fuel ratio over a wide range of air flow rates, and to give manual adjustment of the mixture strength. Essentially, it is a propane pressure regulator and venturi type carburetor. It will operate satisfactorily with propane vapor at about 130 psi pressure (the normal vapor pressure of propane at room temperature). At the rates of propane consumption in a miniature engine, there seems to be no advantage in supplying liquid propane rather than vapor to the pressure reducing valve. In fact, there is a distinct advantage in supplying vapor to the pressure regulator, since the size of the orifice in the venturi can be made much larger, and thus reduce clogging tendencies and make machining easier. A schematic diagram of the carburetor is shown in Figure 32. This carburetor was designed to give a pressure at chamber B equal to atmospheric pressure, or slightly greater or less than atmospheric pressure, as desired. The initial design has a 3/4 in. diaphragm, with a spring having a spring constant of 5.0 lbs per inch. It acts against a tire-type air

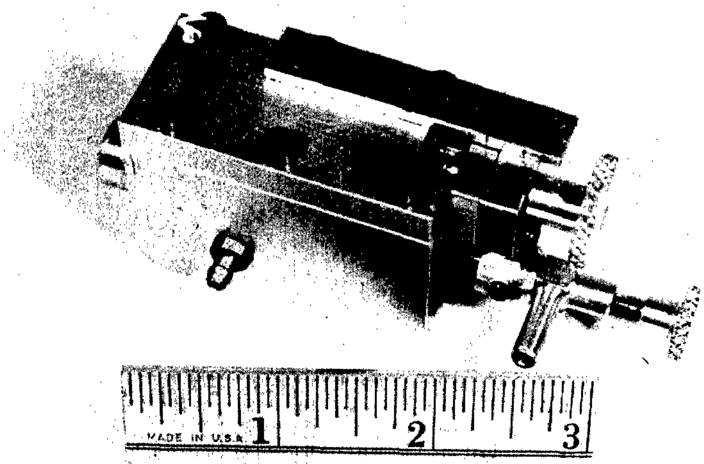


Figure 30. View of Assembled Variable Pressure Experimental Carburetor

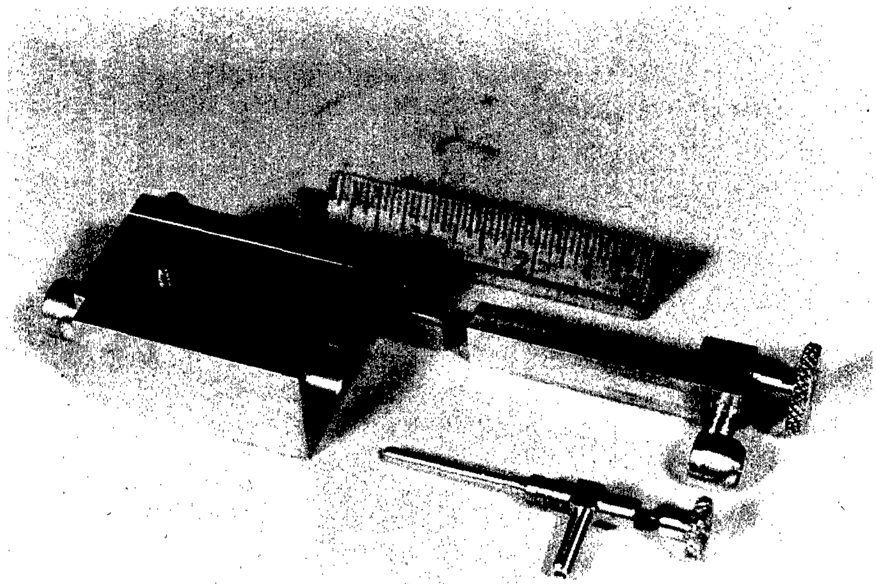
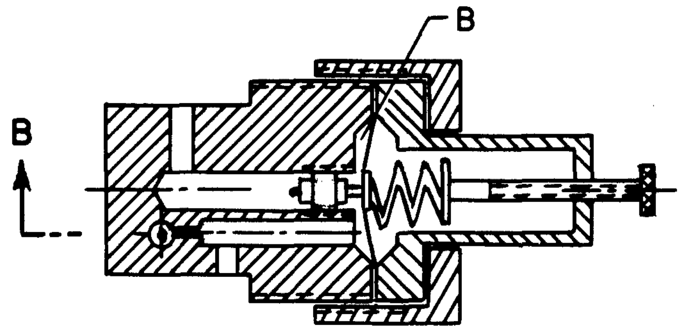
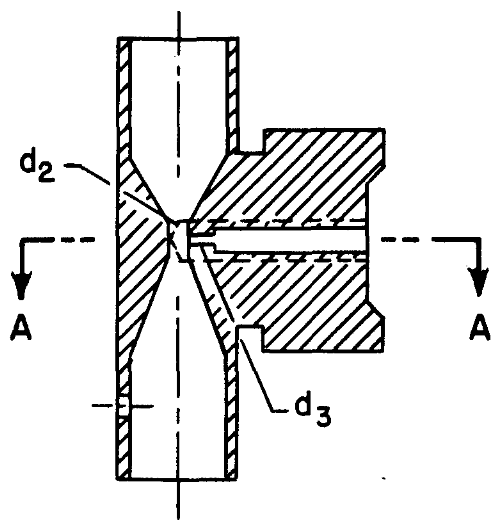


Figure 31. View of Partial Disassembly of Variable Pressure Carburetor Venturi Entrance, Nozzle and Needle-Valve Subassembly, and Nozzle Positioning Subassembly



SECTION. A-A



SECTION B-B

Figure 32. Diagram of Propane Carburetor

check valve, which restricts the propane flow to maintain a constant pressure in chamber B. During tests, a manometer connected at chamber B indicated that uniform propane pressures could be maintained with no visible pressure pulsations. Tests with two different two-cycle engines showed that good engine performance with good starting characteristics is possible with propane. Although the power output has been less for propane than for gasoline or alcohol it is felt that this is the result of improper matching of the venturi throat size and secondary propane pressure to the engine being tested. Figure 35 is a comparison of performance parameters of a two-cycle engine using propane 90-10 mixture of gasoline and oil, or an 80-20 mixture of alcohol and castor oil.

The secondary pressure regulator was redesigned, using the same principles of design as the original model, but using a 1.5 inch diameter diaphragm instead of the original 3/4 in. one (Figure 33). The larger diameter diaphragm permits the use of a stiffer spring. Test results indicate much more uniform regulation of the propane pressure and smoother, less critical engine performance. The primary pressure regulator of the commercial type used in the tests on the original model remained the same. However, a major redesign is required if it is to be used for miniature engine-generator application, as the present commercial size of regulator is nearly as large as a miniature engine.

A pressurized fuel system appears to be the most feasible method of regulating fuel flow considering the requirement of engine operation in any position. Such a system when operating properly affords good control of the fuel flow by either varying the fuel pressure or emitting a small quantity of air into the fuel immediately ahead of the nozzle orifice. Below is a list of components deemed necessary for satisfactory performance:

1. Pressurized fuel tank either by a hand pump or an inert gas from a commercial compressed gas cylinder (carbon dioxide, nitrogen, etc.).

2. Fuel shut-off valve for storage or when engine-generator unit is inoperative for some time.

3. Pressure regulator to decrease the fuel tank pressure and maintain a constant pressure at the fuel metering element.

4. Fuel metering element either of the constriction type or of the flow resistance type.

5. Air-bleed system for minute adjustment of the fuel flow.

6. Venturi to meter the air.

7. Fuel filters at fuel tank filler cap, inlet of fuel pick-up line, and immediately before fuel metering element in carburetor.

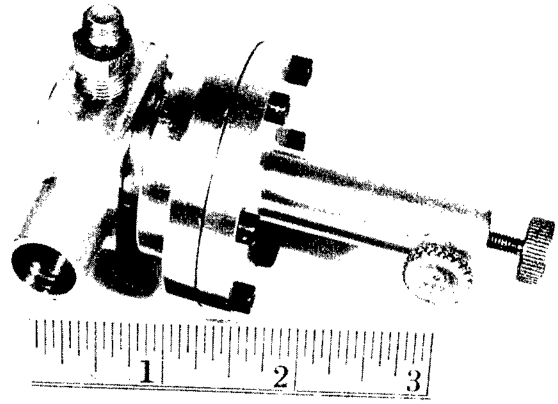


Figure 33. Assembled View of Redesigned Propane Carburetor Showing the 1.5 in. Diaphragm, Fuel Inlet (pipe fitting), Air Inlet and Spring Adjustment

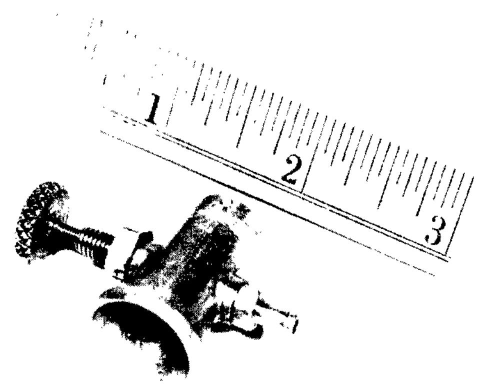


Figure 34. Air Horn and Needle Valve Assembly Used on Commercial Miniature Engines of the Model Aircraft Size

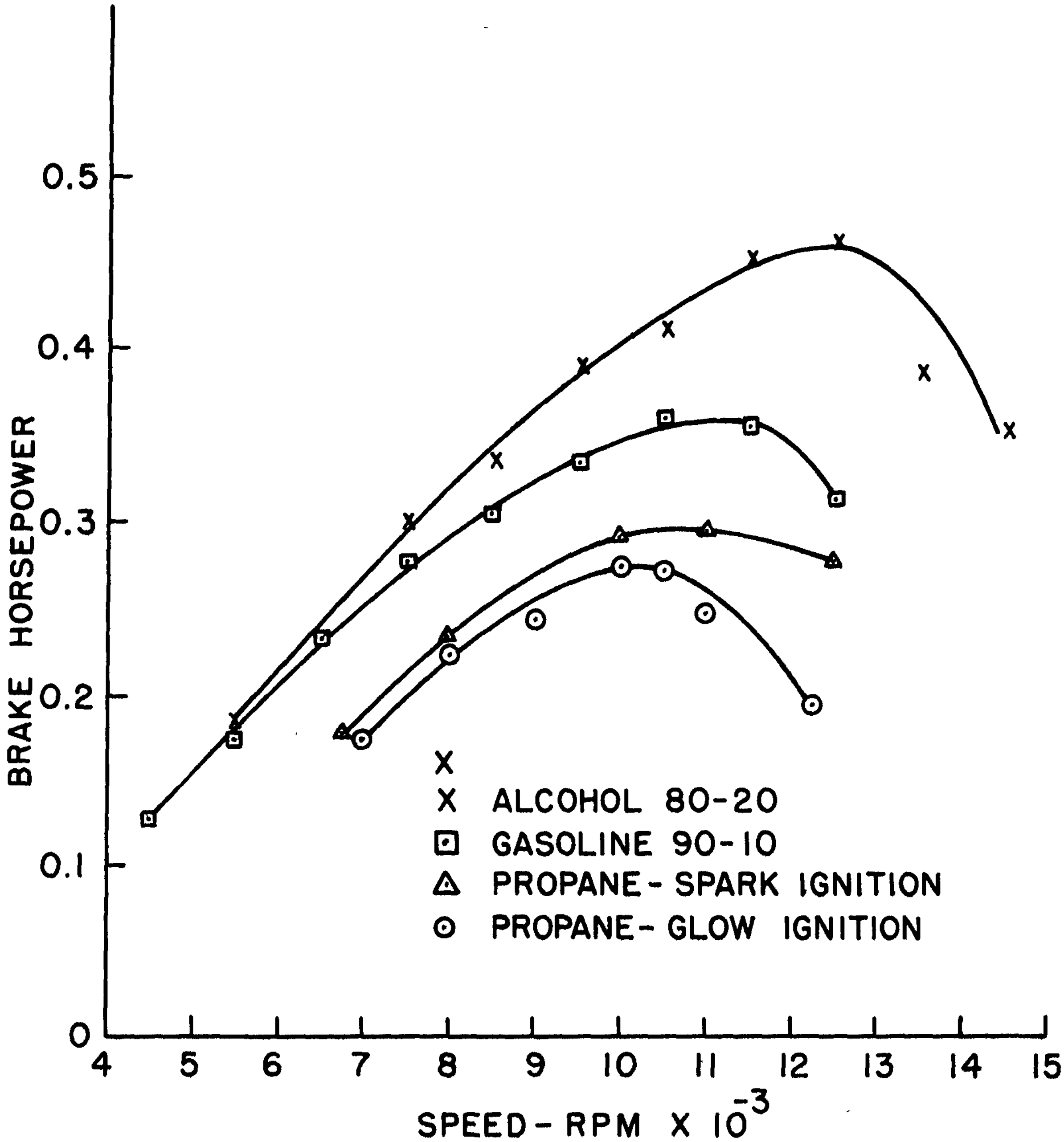


Figure 35. Performance Curves of a Two-Cycle Engine Using Alcohol, Gasoline and Propane

For miniature engine-generators a pressurized fuel system becomes very complex, and as each component is added or incorporated in the design of a simple carburetor to improve its performance, the reliability of the entire unit decreases. Pressure regulators, for example, designed specifically for use with miniature model aircraft engines have proven to be unsatisfactory from a reliability standpoint. The regulators are lightweight and very small in size but because clogging of the minute flow passages and because diaphragm failures has rendered them inoperative after a short running period, considerable redesign is necessary.

Components causing the least trouble are the fuel tank, means of pressurizing fuel tank (viz. a hand pump or compressed inert gas), fuel shut-off valve and filters. A filter of sintered bronze restricting particles of 10 microns or larger works satisfactorily.

Although considerable work was completed during this project on methods of metering fuel none was found to be completely adequate for use with a miniature pressurized system. The very tiny jeweled orifice used as the fuel metering element in a commercial design seemed inadequate due to fouling possibilities. The experimental models of fuel regulators tested were bulky and awkward to handle, but the principles used in their design in conjunction with air-bleed might well provide a satisfactory solution to the problem of fuel control.

While the venturi metering the air seemed trouble-free, it should be kept in mind that too small a throat area will seriously restrict the intake air. However, to obtain a reasonable variation in throat pressure with changes in air flow, the venturi must be something more than a straight pipe.

5. CONTROLS

5.1 INTRODUCTION

This phase of the project was devoted to the study of various means that might be applicable to the control of the engine-generator set. In its broadest sense this would mean the independent or interrelated control of the engine and of the generator to produce the electric energy in a desired form. The word form pertaining to wave shape,frequency, voltage and other generated properties that might vary and thus be controllable.

Considering, then, that the maintaining of a defined electrical output from the generator was the goal to be achieved, and further, considering that this defined output from the generator could be maintained without undue complications by electrical designers provided the generator speed remained constant; the engine-generator set control problem was reduced to the determination of factors that, when varied, would offset load changes and impose constant speed to the generator. It was felt that this constant speed might be obtained by either varying engine control factors or by establishing some form of control load that would, at least as far as the engine-generator set could perceive, maintain a constant load.

With the problem now having been reduced, in part, to the maintaining of a constant engine speed under conditions of varying load, an investigation was conducted to determine how this control might be achieved on either a two-cycle or a four-cycle engine. Many of the control variables considered were evaluated by tests on specific engines, and the results of these tests plus additional considerations are presented in the remainder of this section.

The control of the actual generator output was not considered under this phase of the project, for it was felt that if this constant speed could be achieved, the design of the generator controls would be reduced to a point where present theory and practice could be applied without additional contributions. Therefore anything that might be pertinent to generator control design alone will be found in the reports considering alternating and direct current generation.

5.2 ENGINE CONTROL

5.2.1 General

The object of these studies was to investigate a method of control, the effectiveness of this method, and the economics involved in controlling the engine speed under load variation with the method. The study considered both two-cycle and four-cycle engines and the following list of variables was compiled for investigation:

1) Two-Cycle Engine
   a) Throttling of inlet air-fuel.
   b) Variable crankcase compression ratio.
   c) Variable exhaust back pressure.
   d) Variable timing on spark ignition.
   e) Variable compression ratio.
   f) Variable timing of rotary valve on crankcase.
   g) Variable mixture strength admitted to the cylinder.

2) Four-Cycle Engine
   a) Throttling of inlet air-fuel.
   b) Variable exhaust back pressure.
   c) Variable spark plug timing.
   d) Variable compression ratio.
   e) Variable valve timing.

All of the variables listed were examined by actual engine performance tests except for three. The three control methods not tested were variable compression ratio on the two-cycle engine and variable compression ratio plus variable valve timing on the four-cycle engine.

Variable compression ratio on either engine was not attempted because of the complications that seemed inevitable to produce this variation. It was realized that variable compression could be achieved by shimming between the head and block, but this would have provided "stepwise" variation that, in addition, would have necessitated a shut down between setting. This type of test would demonstrate the influence of compression ratio on performance, but certainly would not be acceptable for controlling the engine during continuous operation at varying loads. Therefore, such tests were not conducted.

Variable valve timing on the four-cycle engine was not attempted for a similar reason. Since the valve cam shaft is normally gear or chain driven by the crankshaft, timing can be varied in steps but to produce a continuous variation in the timing would take a rather complex mechanism, which certainly would not be consistent with miniaturization. The conclusion was therefore drawn that tests on such a variable would not be in order and so none were conducted.

5.2.2 Two-Cycle Engine Tests

5.2.2.1 Throttling of inlet air-fuel. The throttling of the inlet air-fuel was accomplished by means of a standard, float type, Tilletson carburetor. The carburetor was located ahead of the rotary valve at the crankcase and the position of the butterfly valve was controlled manually. To accomplish the investigation with as little complication as possible the exact position of the butterfly was not indicated, but only relative positions were used. The testing was conducted by varying the load and maintaining the engine speed constant by varying the carburetor setting.

The curves of Figure 36 and 37 show the variations of brake horsepower, total fuel consumption, and brake-specific-fuel consumption versus relative carburetor setting that were obtained from the tests at 8,000, 10,000 and 12,000 rpm. The power variation of from 0.252 to 0.370 brake horsepower, or 38 per cent, was the greatest variation obtained and it occurred at 10,000 rpm. Power variations of about 14.7 and 21.5 per cent were obtained at 12,000 and 8,000 rpm, respectively. The fact that the 8,000 and 12,000 rpm curves peak and then decrease again can be attributed to the carburetor not being matched to the engine. These are not typical curves for internal combustion engines operating at constant speed with variable carburetor settings. All the curves should be similar to the 10,000 rpm curve, that shows tendencies that are typical.

The total fuel consumption curves show very little variation with power output or carburetor setting. Such results are typical of data obtained from most miniature engine tests; total fuel consumed in lb/hr is more a function of a particular engine than it is of the load or the rpm. These results indicate also that the carburetor was not matched with the engine, for with the total fuel consumption remaining constant an opening of the throttle leads to a leaning of the mixture. A proper matching of carburetor to engine was not attempted under the control phase, but was considered and is discussed under the carburetion phase. Any designer considering the use of a carburetor for control should investigate its range of applicability when the particular carburetor-engine combination is chosen.

To carry the investigation of throttling inlet air-fuel further, a second set of tests were conducted. The Tilletson carburetor was replaced with a combination of butterfly valve and needle valve located in a venturi tube, Figure 38.

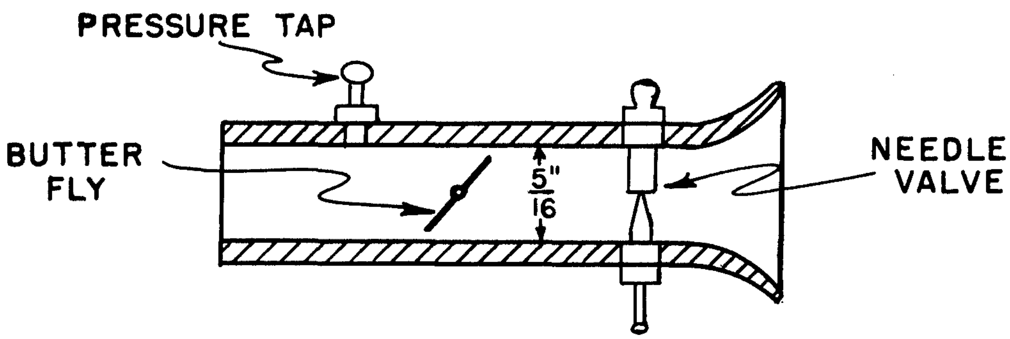


Figure 38. Venturi Tube for Throttling

This arrangement permitted the variation to be made without a leaning of the mixture. A pressure tap was also included to give an indication of the variation in manifold pressure.

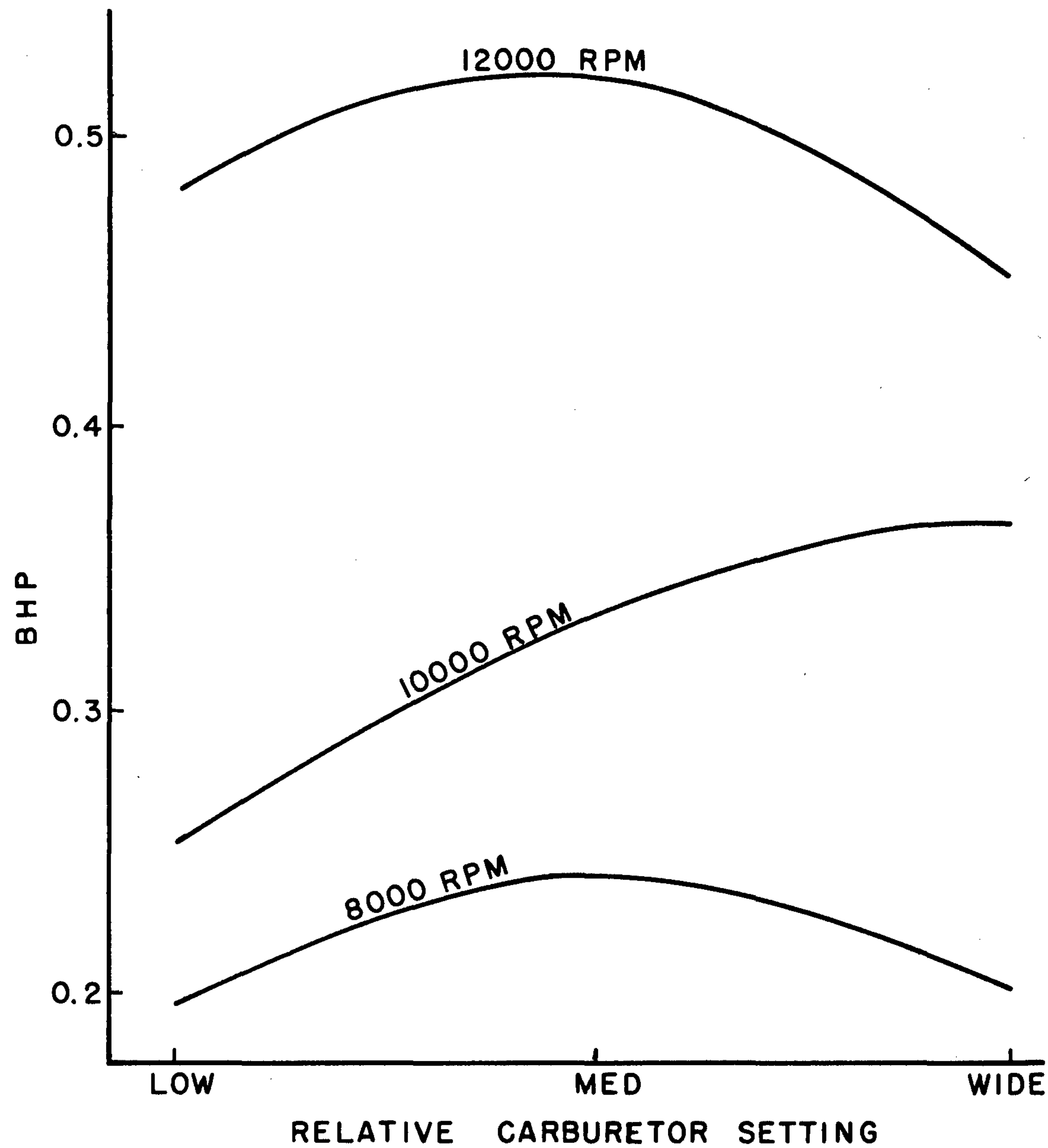


Figure 36. Brake Horsepower vs. Carburetor Setting for Mark II Engine Operating at Constant Speed

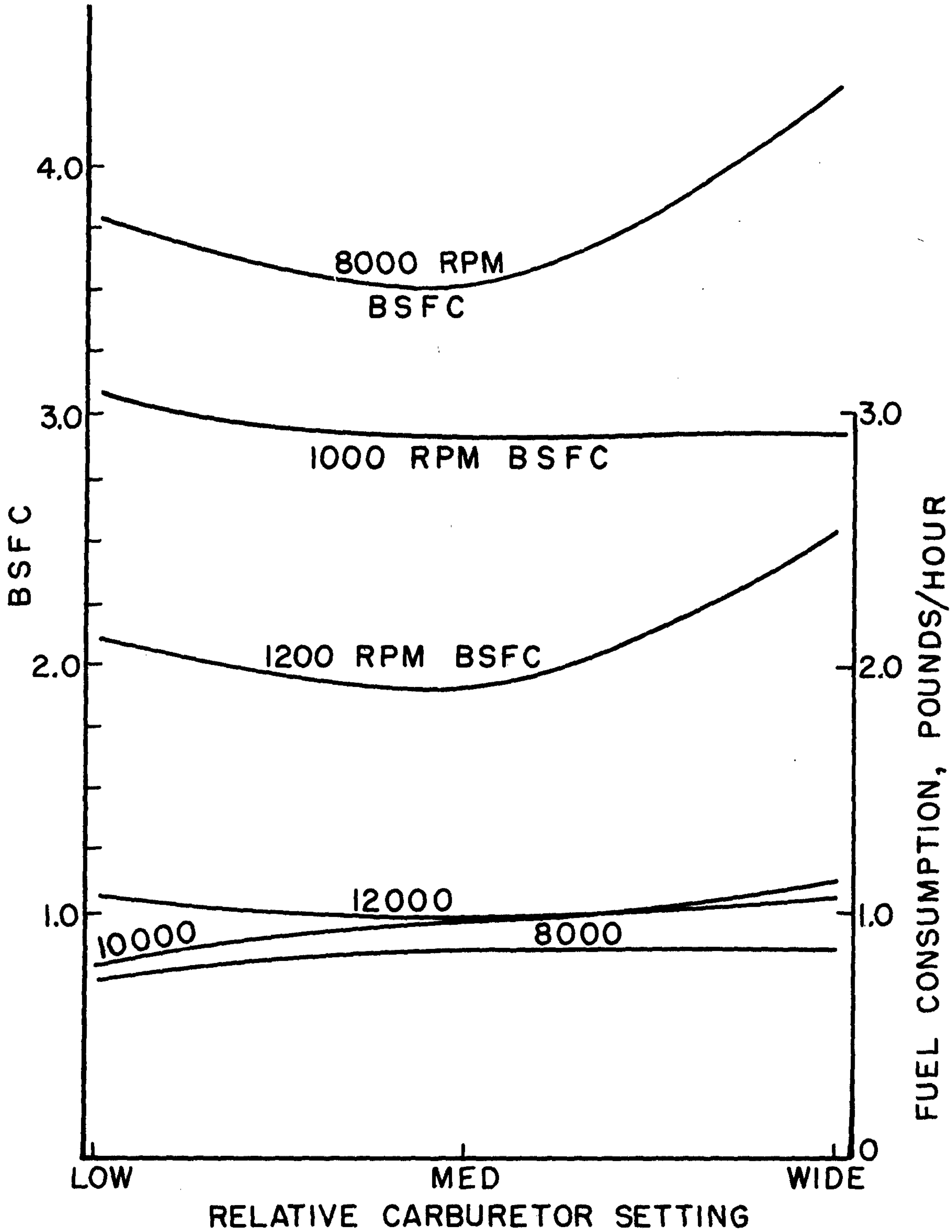


Figure 37. Total and Specific Fuel Consumption vs. Carburetor Setting for Mark II Engine

Figure 39 shows the results of these tests and gives indications that, with a properly matched carburetor, it should be possible to obtain a typical power variation with throttle position. The power variations obtained were 80 per cent at 12,000 rpm, 67 per cent at 10,000 rpm, and 79 per cent at 8,000 rpm. Since the variation is typical there is little reason to assume that control by this means, with a matched carburetor, could not be achieved.

5.2.2.2 Variable crankcase compression ratio. A piston-cylinder type device was utilized that permitted the continuous variation of the crankcase compression ratio. It was possible with this device to vary the crankcase compression ratio while the engine was operating and thus to determine directly the influence of this variable. The piston was positioned in its cylinder by means of a nut and screw arrangement and held in the set position with a lock nut.

The investigation was carried out by operating the engine at a given speed and under a given load. The crankcase compression was then varied over its entire range to determine the variation of engine performance. The next step would have been to vary the load and offset this load variation with a change in crankcase compression ratio, however, since the engine exhibited no response to a variation of crankcase compression, Figure 40, this test never advanced beyond one power setting for each speed.

In reviewing the curves presented in Report No. 8*, which indicated that power output varied with crankcase compression, it was discovered that along with the change in the crankcase compression ratio for these tests there was also a change in the timing of the rotary valve. This latter introduces a second variable, namely, timing of the admission of air-fuel into the crankcase which also influences the power output. The fact that it does influence the power output is reported under the title, Variable Timing of Rotary Valve on Crankcase, later in this section.

The final conclusion to be drawn is that the crankcase compression ratio apparently has no influence on the power output of the two-cycle engine - it being felt that the tests conducted under the control phase were more indicative of truly crankcase compression variation. This conclusion is of more significance to the over-all engine design than to the control phase for it means that the designer need not be too concerned with the crankcase compression ratio and can therefore enjoy greater freedom in design.

5.2.2.3 Variable exhaust back pressure. Since the exhaust from the engines was removed through a duct by means of an induced draft fan to overcome duct loss, it was a simple matter to install a butterfly type valve in the exhaust from the engine and thereby vary the pressure at the exhaust from the engine. To reduce complication, the magnitude of the back pressure was not measured but merely inferred from the position of the butterfly valve. It was possible with this arrangement to vary the exhaust back pressure while the engine was operating, therefore the tests were conducted again by varying the load and maintaining a constant speed by changing the pressure at the exhaust.

* Quarterly report on Contract No. AF 18(600)-192

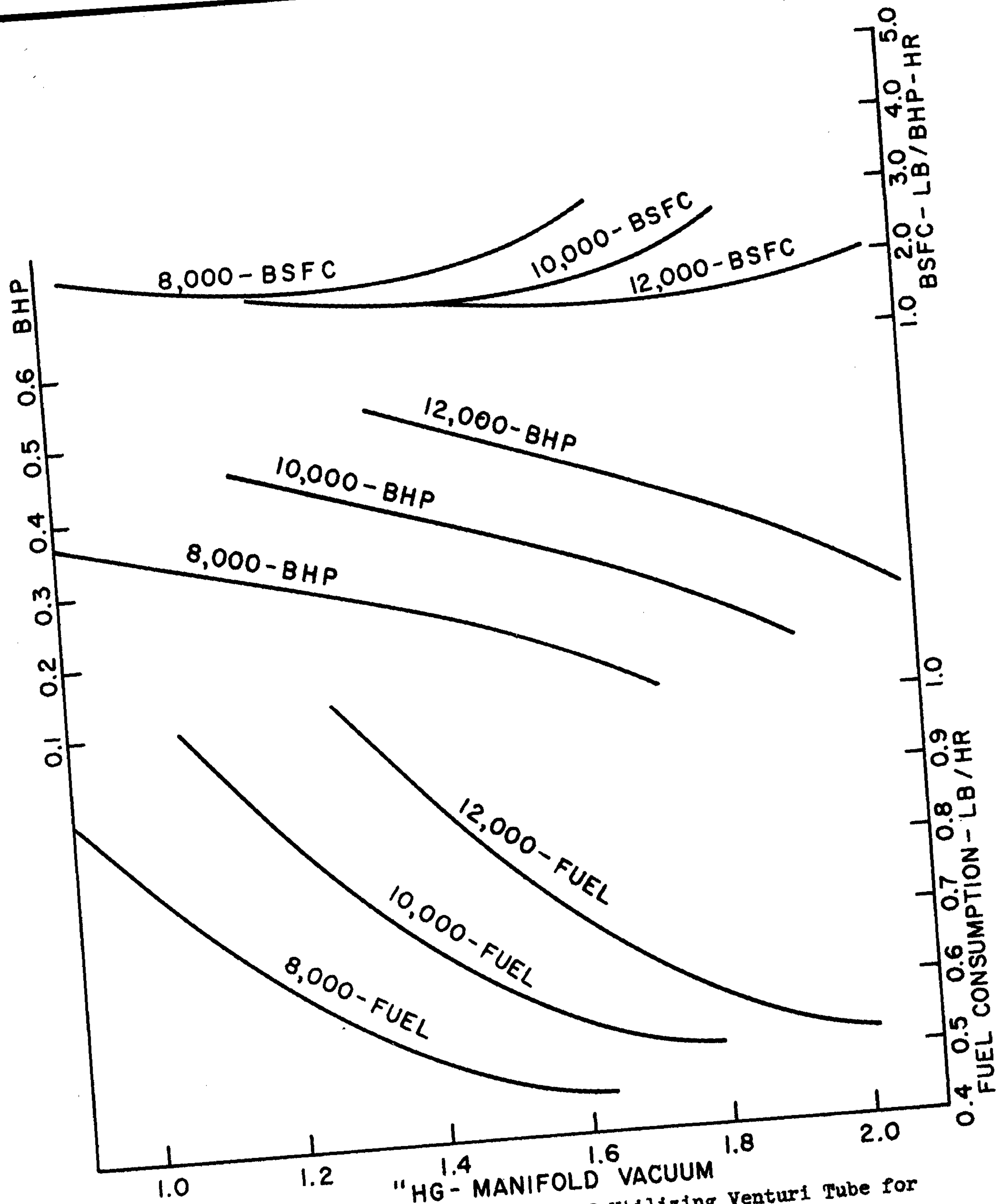


Figure 39. Performance Curves for Mark II Utilizing Venturi Tube for Throttling

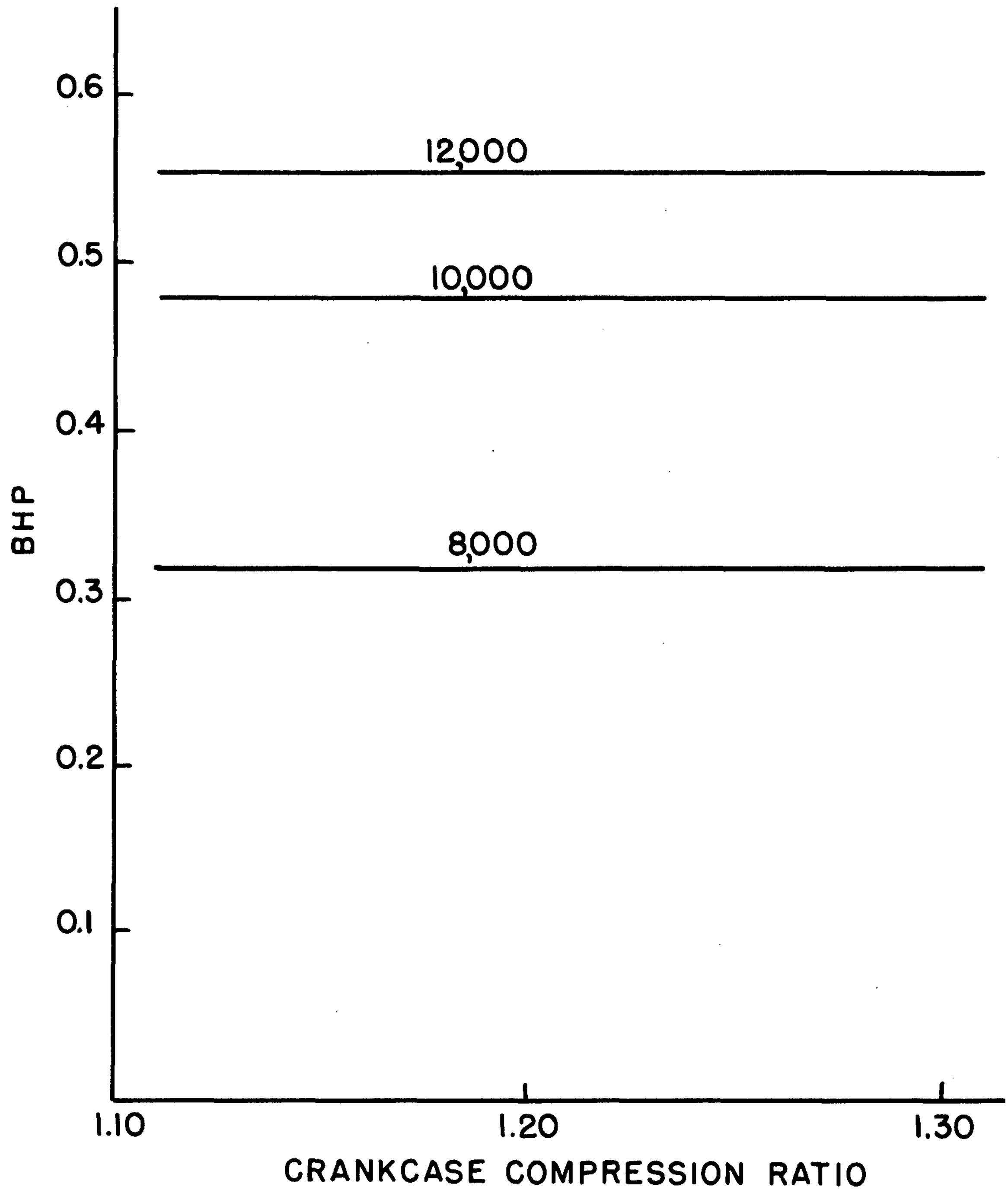


Figure 40. Brake-Horsepower Curves for Mark II Showing Effect of Crankcase Compression Ratio on Power

Since the curves presented in Report 15*, Fig. 2, left much to be desired, the tests on variable exhaust back pressure were rerun using a new butterfly valve. Also, due to the difficulty in accurately determining the position of the butterfly valve, a pressure tap was incorporated ahead of the valve, i.e., between the engine exhaust port and the butterfly valve. The plots of brake horsepower, fuel consumption, and specific fuel consumption were then made versus exhaust pressure.

The engine was operated utilizing both spark ignition with a 90-10 aviation gas and oil mixture and glow plug ignition using an 80-20 alcohol and castor oil mixture. The fact that both negative and positive pressures were obtained is explained when it is remembered that an induced fan was incorporated in the exhaust system and its speed could be varied electrically.

Figure 41 shows the variation of brake horsepower, total fuel consumption, and specific fuel consumption versus exhaust pressure with spark ignition. Considering speeds of 8,000 rpm, 10,000 rpm, and 12,000 rpm, it can be seen that there is virtually no variation in power below an exhaust pressure of 0.4 inches of mercury. The same lack of variation can also be seen for the total and specific fuel consumption curves. Above 0.4 inches of mercury the total fuel consumption still does not vary, but, due to the decrease in brake horsepower, the specific fuel consumption rises.

At 12,000 rpm the power decreased from 0.41 horsepower to 0.32 horsepower or a change of 24%. The greatest change in power output was from 0.4 horsepower to 0.21 horsepower or about 62 per cent, at 10,000 rpm. These values, since they are the result of improved testing procedures, should be weighted more than earlier tests, and as such, the conclusion now would be that the influence of exhaust back pressure is more typical than indicated in the earlier tests.

Figure 42 shows the variation of brake horsepower, total fuel consumption, and specific fuel consumption versus exhaust pressure using a glow plug. The curves have been discontinued at a back pressure of zero since no further change in power was observed. This is similar to the variation shown in Figure 41 at pressures below 0.4 inches of mercury. The greatest power variation, percentage wise, was at 12,000 rpm while the greatest change in power 0.235 to 0.618 horsepower occurred at 14,000 rpm.

5.2.2.4 Variable timing on spark ignition. The test facilities were such that it was possible to vary the spark timing while the engine continued to operate. Consequently, the tests were once more conducted by varying the load and maintaining the speed constant by adjusting the variable, which in this case was the spark plug timing.

The curves of Figure 43 and 44 show the variation of brake horsepower, total fuel consumption, and brake-specific-fuel consumption versus spark plug timing for speeds of 12,000, 10,000, 8,750, and 7,500 rpm. All the curves indicate that the engine was materially influenced by the timing of the spark, and a comparison will quickly show that this variable had the most influence of all the variables attempted on the two-cycle engine.

* Quarterly report on Contract No. AF 18(600)-192

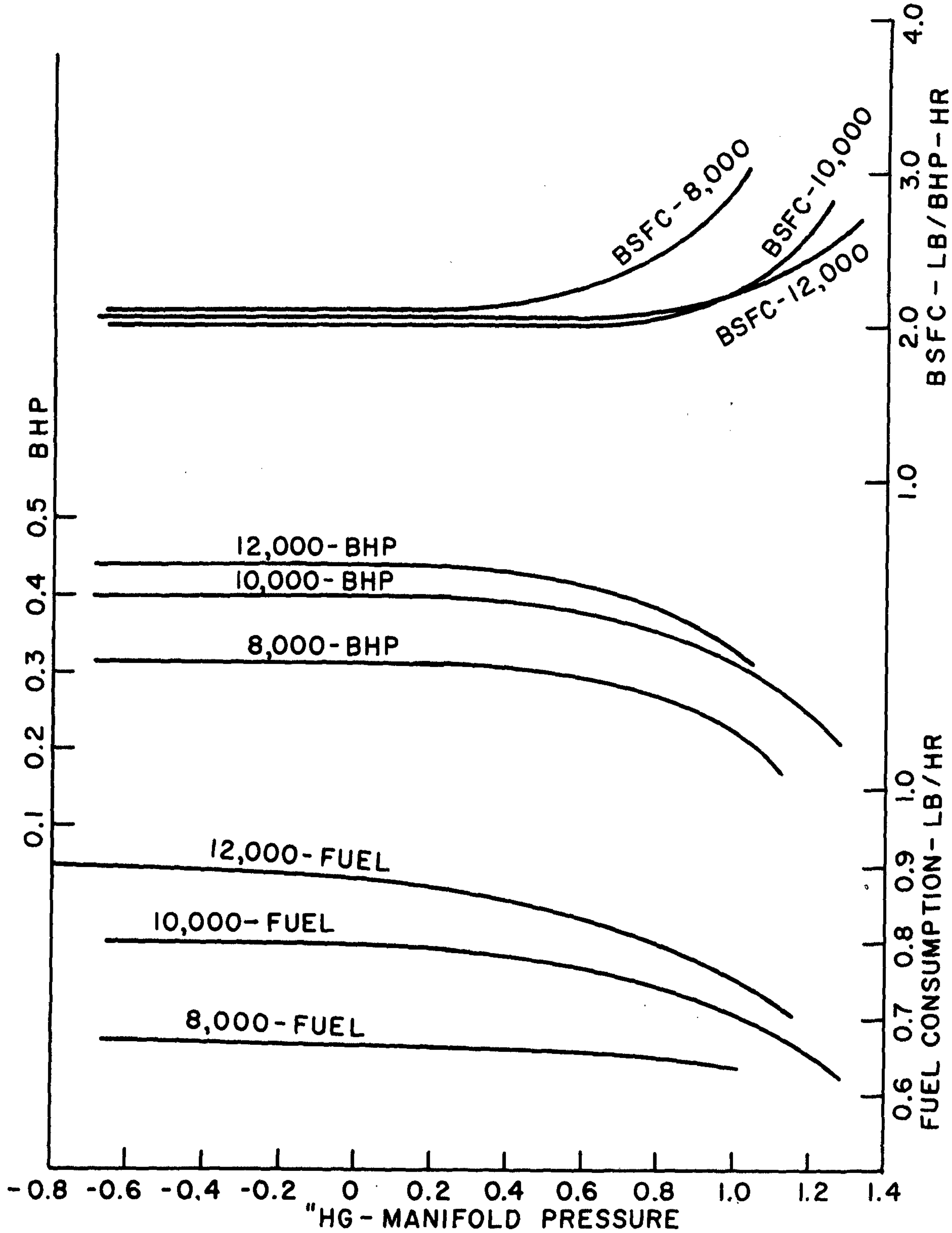


Figure 41. Performance Curves for Mark II Showing Effect of Exhaust Back Pressure When Spark Ignition is Utilized

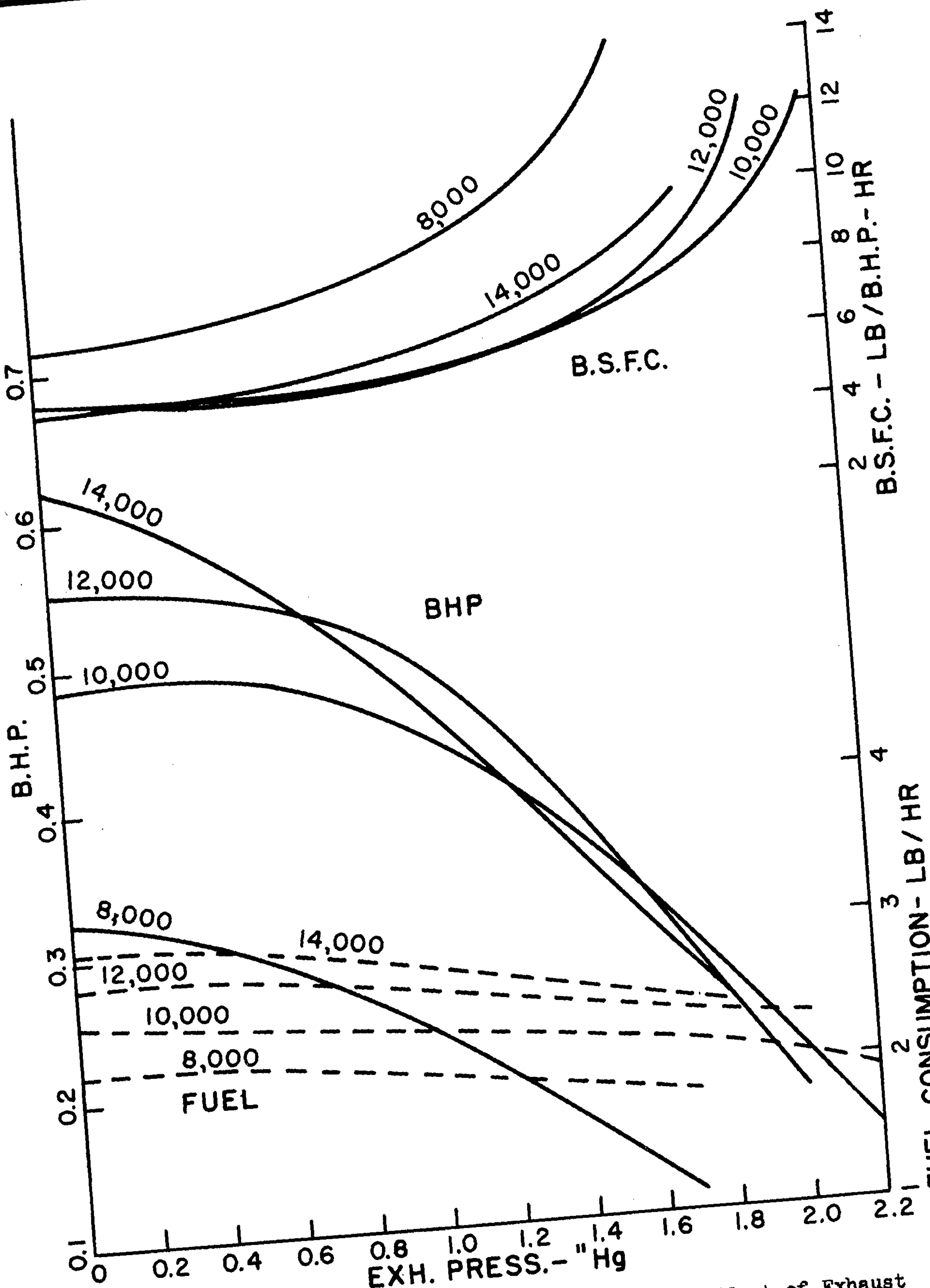


Figure 42. Performance Curve for Mark II Showing Effect of Exhaust Back Pressure With Glow Plug Ignition

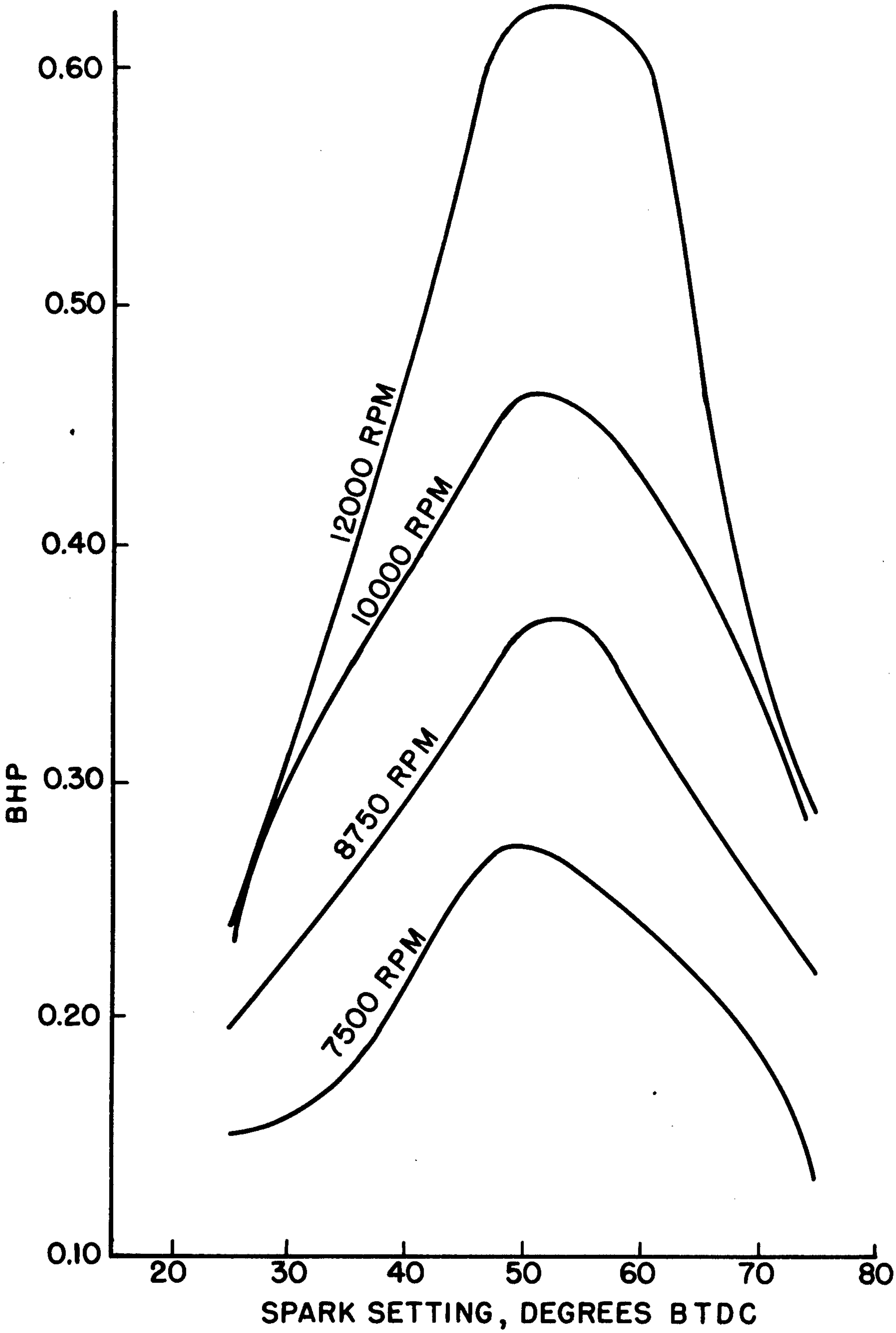


Figure 43. Brake Horsepower vs. Spark Setting for Mark II Engine Operating at Constant Speed

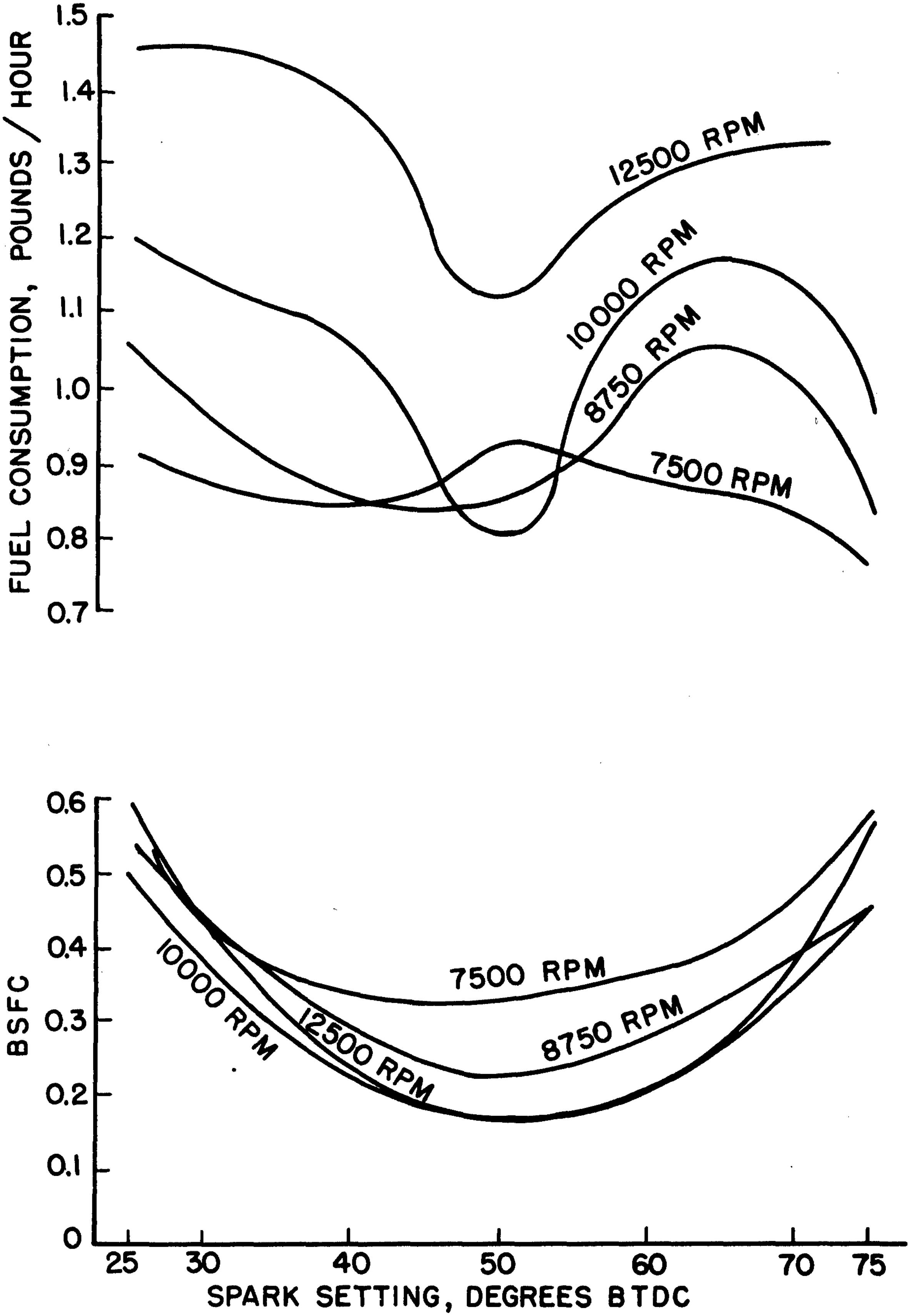


Figure 44. Total and Specific Fuel Consumption vs. Spark Setting for Mark II Engine Operating at Constant Speed

Considering the power curves, the 12,000 rpm test indicated the greatest variation of brake horsepower. The variation of from 0.242 to 0.625 horsepower represents about 88.5 per cent based on the average value. The 10,000, 8,750, and 7,500 rpm tests indicate variations of about 65.5, 60, and 70 per cent, respectively. All of the curves peaked and for all practical purposes, can be seen to be symmetrical with respect to the peak setting. The peak power varied from speed to speed as is typical with all internal combustion engines.

The total fuel consumption curves, although a little peculiar in shape, give variation of 26 per cent at 12,500 rpm, 39 per cent at 10,000 rpm, 22.3 per cent at 8,750 rpm and 18.8 per cent at 7,500 rpm. The brake-specific-fuel consumption curves take on a very orderly appearance and show their minimum values near the peak power output.

In general the results of the variable spark timing tests were in accord with those obtained when larger engines are subjected to the same type of testing.

5.2.2.5 Variable timing of rotary valve on crankcase. Variable timing of the rotary valve was accomplished with a movable section set into the crankcase rather than by repositioning the rotary valve on the crankshaft. This method was chosen since it permitted the change to be made while the engine was in operation.

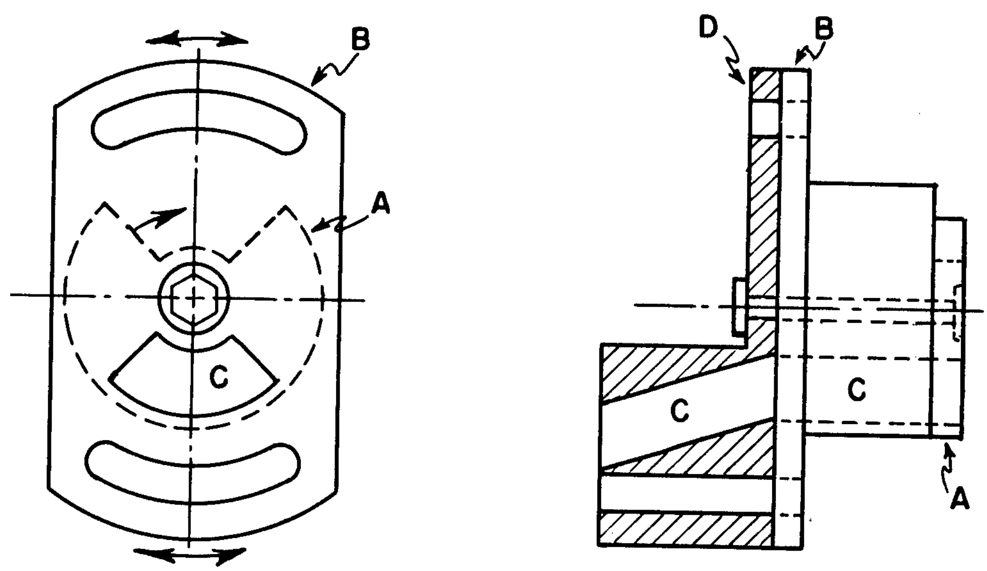


Figure 45. Method for Varying Rotary Valve Timing

Referring to Figure 45 the rotary valve A is driven by the crankshaft, B is the movable section, and D is a stationary section. Inlet passage C is

continuous through parts D and B. With the slots and passage in B it is possible to reposition B and thus change the timing of the opening into the crankcase.

As mentioned under the section on Variable Crankcase Compression Ratio when the rotary valve timing is changed a change is also made in the crankcase compression ratio; however, since the tests on crankcase compression ratio indicated that this variable had no influence, no efforts were made to allow for the crankcase compression change. Further, since the movable section could be repositioned while the engine was in operation, the tests were conducted as in the other tests, by varying the load and maintaining a constant speed by compensating with the variable.

The curves on Figure 46 show the variation of brake horsepower and total fuel consumption versus time of rotary valve opening in degrees after top dead center for speeds of 12,500, 11,000, and 9,500 rpm. The power curves indicate that the maximum variation was obtained at 12,500 rpm, where the brake horsepower varied from 0.7 to 0.585 horsepower, or about 17.5 per cent based on the average value. The total fuel consumption variation was of the order of 7 per cent.

Considering the limited variations and the complications involved in attaining the variations, it would seem questionable that this method should be considered for control. Complications arise in fabricating the crankcase and the movable section of crankcase so that interference will not be encountered. There is also the problem of controlling the position of the movable section as well as guarding against leakage.

5.2.2.6 Variable mixture strength admitted to the cylinder. While the leaning or richening of the mixture normally produces an effect on the power output of an engine, miniature engines, because of their sensitivity to fuel, respond to a mixture change by either running or not running. An examination of the results from other tests showed that the total fuel consumption remains essentially constant and any deviation of any magnitude from this amount of fuel results in a stopping of the engine.

5.2.3 Four-Cycle Tests

5.2.3.1 Throttling of inlet air-fuel. A Thompson carburetor was used to accomplish the throttling of the inlet air-fuel mixture. The tests were again conducted by varying the load and attempting to maintain a constant speed by readjusting the throttle setting on the carburetor. Since this phase of the project was not directly concerned with complete engine performance the exact setting of the throttle valve for each load was not recorded but relative values were observed.

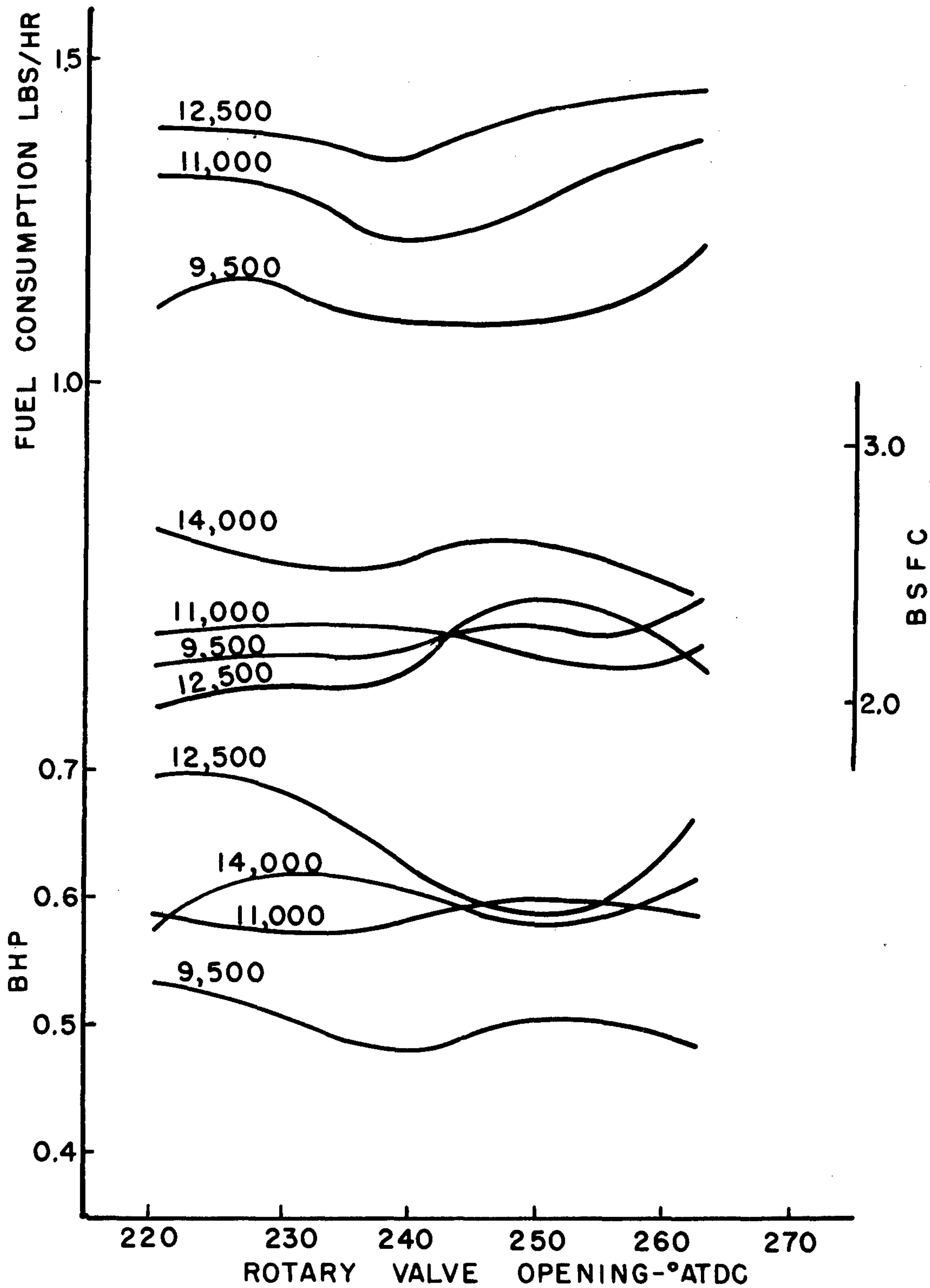


Figure 46. Performance Curves for Mark II, Two-Cycle Engine Showing Effect of Rotary Valve Setting

The curves shown on Figure 47 show the variation of power, total fuel consumption, and brake-specific-fuel consumption versus relative carburetor setting. The total fuel consumption curve shows clearly that the low carburetor setting was the one using minimum air-fuel or a condition where in the throttle valve was well closed, approaching idle. The 8,000 rpm curve shows a variation of 40 per cent in total fuel consumed. The other speeds show a lesser variation, however, the curve of most interest is the 8,000 rpm since this speed represents a desirable design speed for the generators.

The power curve at 8,000 rpm shows a variation of 30.5 per cent in its change from 0.337 horsepower at a low setting to 0.458 horsepower at a medium setting. This curve further exhibits a peculiarity in that as the carburetor is opened wider the power falls off. The curves at 6,000 and 7,000 rpm do not exhibit this tendency but represent what is a normal trend in engines with a throttle variation. The only possible explanation of this curve seems to be that the carburetor used was not matched to the engine. No additional tests were performed, as they were with the two-cycle engine, for it is felt that with a properly matched carburetor the results would be as predicted from the performance of larger engines.

5.2.3.2 Variable exhaust back pressure. The variation in exhaust back pressure was accomplished by varying the angle of a butterfly type valve in the exhaust ducting from the engine. As in previous control tests with variable back pressure the magnitude of the back pressure was not measured but the position of the butterfly was noted to give a relative indication of the back pressure.

The curves on Figure 48 indicate the variation in brake-horsepower with a variation in back pressure. The back pressure variation is depicted by the setting of the exhaust butterfly with zero degrees representing a condition of no back pressure other than that inflicted by the exhaust ducting itself. The higher values, in degrees, indicate the number of degrees the butterfly was closed from the horizontal, and thus, the relative increase in back pressure is shown.

The curve for 10,000 rpm shows a variation in brake horsepower from 0.504 horsepower to 0.444 horsepower or a variation of 12.7 per cent based on the average. The curve for 8,000 rpm shows a variation of only 6.9 per cent while the 6,000 rpm curve indicates a 9.3 per cent variation. The 6,000 and 8,000 rpm curves indicate a decrease in power as soon as the back pressure increases. This was interpreted to mean that as the pressure at the exhaust was increased a correspondingly smaller charge of fuel-air mixture was able to enter the cylinder. This, of course, appears logical since the flow of air-fuel into the cylinder is governed, in part, by the pressure difference between a point in the intake manifold and a point in the cylinder. The higher exhaust pressure would require a higher cylinder pressure to move the products of combustion out of the cylinder, and this higher cylinder pressure would create a smaller pressure differential with which to force the new charge into the cylinder. The behavior of the 10,000 rpm curve, i.e., the increase in power output before the decrease, was thought to stem from the possibility that at the higher speed, and

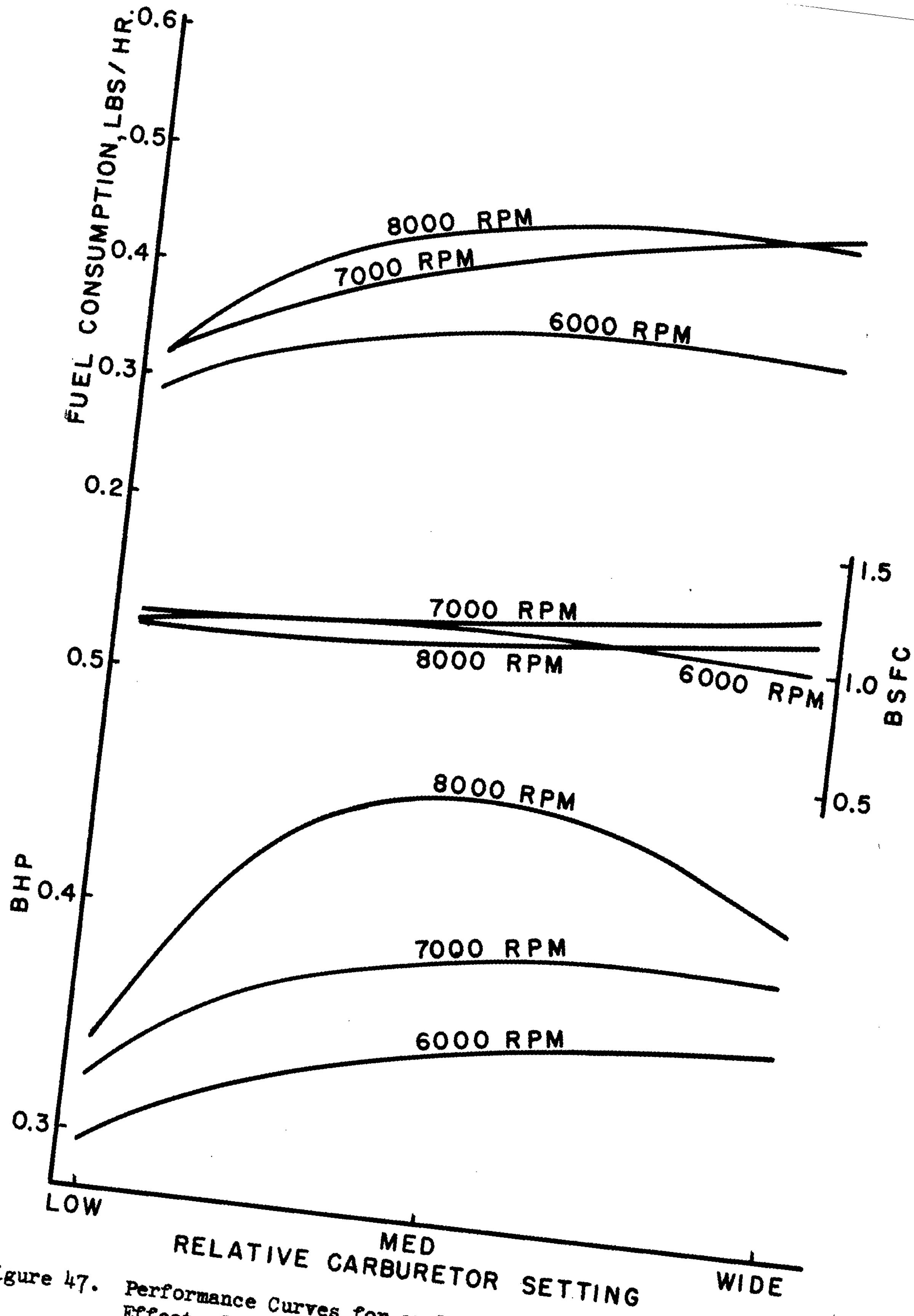


Figure 47. Performance Curves for an L-Head, Four-Cycle Engine, Showing Effect of Carburetor Setting

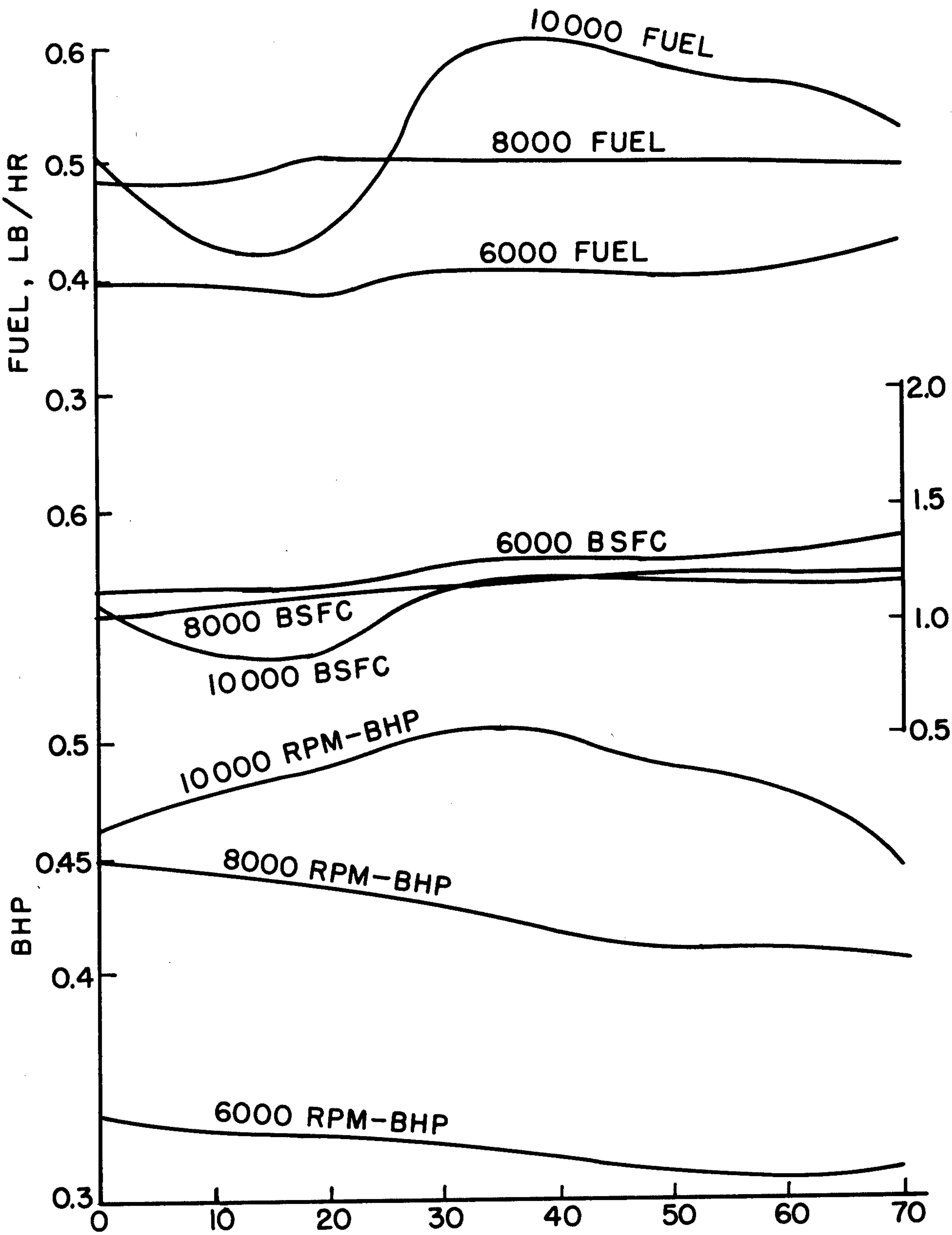


Figure 48. Performance Curves for an L-Head, Four-Cycle Engine, Showing Effect of Back Pressure Variation

with little or no additional exhaust pressure, some of the new air-fuel charge was carried through the cylinder. Therefore, for a while the increase in exhaust pressure actually provided the engine with a better combustible mixture by reducing this blow through. After the peak power, however, it was assumed that the explanation of power decrease fell into the same line of reasoning that was used on the 6,000 and 8,000 rpm.curves.

The total fuel consumption and brake specific fuel consumption curves indicate that the change in exhaust pressure produced virtually no change in fuel consumption, which did not appear unusual since the power variation was so small.

In comparison to the other methods attempted for control of the four-cycle engine, variable back pressure on the engine appears to hold the least promise. As such it would be doubtful if this should be considered as a possible means for package control.

5.2.3.3 Variable spark timing. Since the test facilities permitted spark timing to be varied while the engine was operating, the tests were again conducted by attempting to maintain constant speed while the load and timing were varied. The timing was varied manually and no attempts were made to devise any form of automatic actuating mechanism to vary the timing. This philosophy was in keeping with that used in other portions of the control phase, that is, the method was explored and not the mechanics of operating a control mechanism.

The curves on Figure 49 show the variation in brake-horsepower, brake-specific-fuel consumption, and total fuel consumption for four different speeds versus spark timing in degrees before-top-dead-center. The 8,000 rpm curve shows a variation of about 48 per cent, from 0.323 to 0.530 horsepower, which is the greatest variation of any of the speeds investigated. The power variation for the other speeds was about 36.5, 32.5, and 39 per cent for 9,000, 7,000, and 6,000 rpm, respectively. The fact that the greatest variation occurs at 8,000 rpm is of particular interest since this is a desirable design speed for the generators. Further, the steepness of the 8,000 rpm curve between 0 degrees and 20 degrees gives indications that this would be a good range for control since there was a reasonable variation in power for each variation in spark advance. The power variation curves are similar to those obtained on any spark-ignition engine, except it is interesting to note that power was relatively insensitive to spark timing at early spark setting.

The total fuel consumption curve for 8,000 rpm shows very little variation over the entire spark advance range and practically no change in the control range. Obviously, the brake-specific-fuel-consumption curve then must show a variation and the minimum specific fuel consumption comes at the point of maximum power output, which would be expected with a constant total fuel consumption. The fact that the total fuel consumption does not vary in the control range is of interest, for this means that regardless of the power delivered by the engine, the same amount of fuel would be required for the same length of time of operation. Therefore, operation at partial power would not permit any weight saving over continuous operation at rated power.

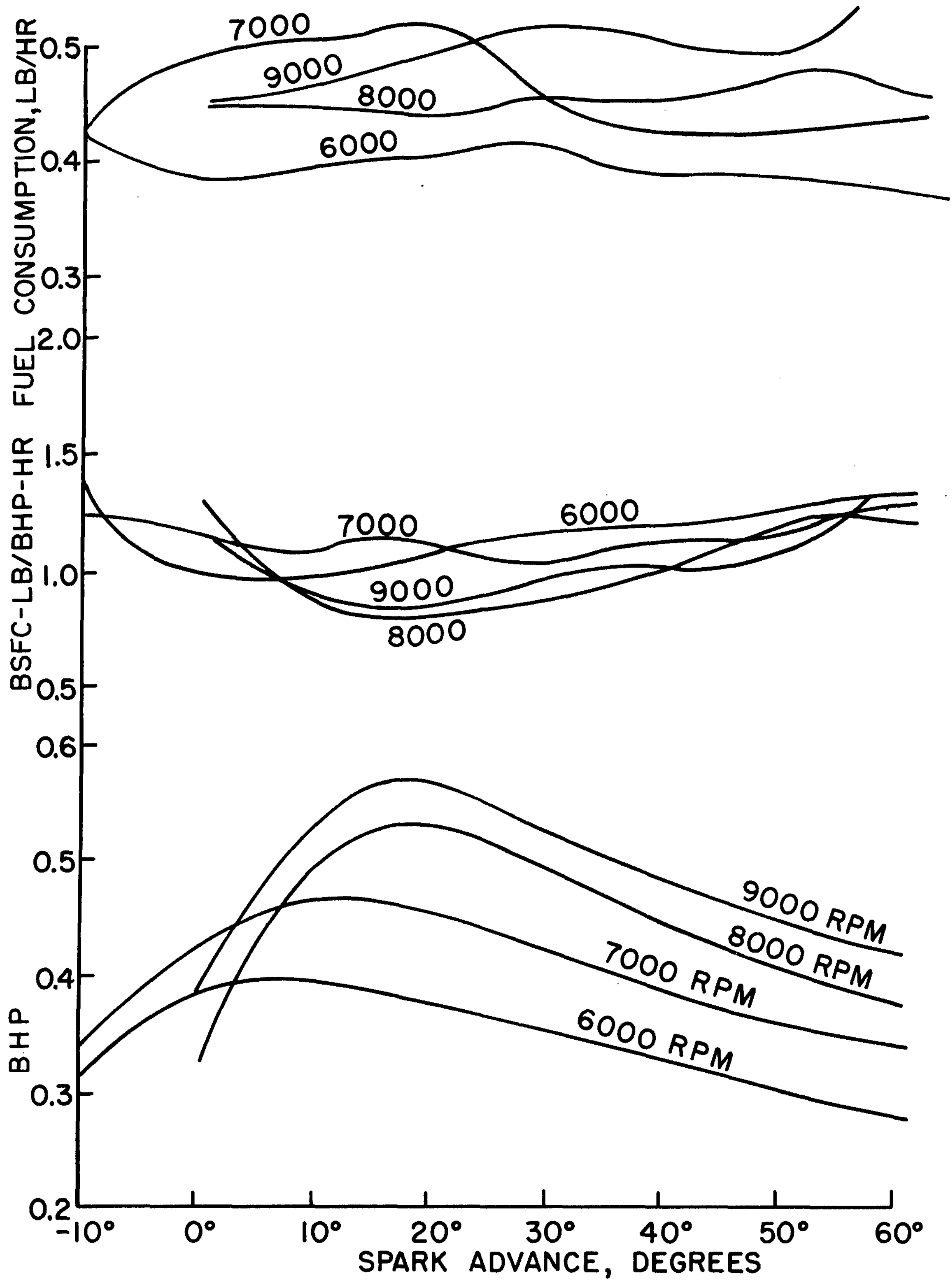


Figure 49. Performance Curves for an L-Head, Four-Cycle Engine Showing Effect of Spark Setting

5.3 PACKAGE CONTROL WITH ENGINE

Although all of the engine control variables were not attempted on an actual engine-generator package, there is no reason to assume that the results would be any different from those presented here and taken from the engine-dynamometer tests. It should be remembered, however, that these results should be considered as trends rather than the results of repeated tests for, as in the general performance testing, reproducibility of actual values left much to be desired. The trends, however, were reproducible and therefore were included.

The tests indicate that various of these methods presented could be used for control over desired ranges. The only variables exhibiting no influence on the engine and thus not suited for control were crankcase compression and variable mixture strength. The variation of exhaust back pressure gave the greatest range of control on the two-cycle engine when using the glow plug; however, the variation of spark timing or throttling of inlet air-fuel also gave suitable results. It must be remembered that a matched carburetor is needed to accomplish the throttling control successfully and the development of this carburetor could be time consuming.

Of the three variables actually tested on the four-cycle engine, only exhaust back pressure failed to give a fair degree of control. Both throttling of inlet air-fuel and spark timing gave power variations of 30 per cent or slightly greater. It should again be remembered, however, that a wider control range from throttling air-fuel would be expected if the carburetor were better matched to the engine.

A general conclusion that might be reached is that none of the variables tested effected the engine performance as greatly as these same variables do when applied to larger engines.

5.4 PACKAGE CONTROL THROUGH LOAD

Another method of package control would be one that permitted no load variation for the engine-generator combination. That is, in so far as the engine and the generator are concerned, there would be no change in the load regardless of how much the actual load varied.

To accomplish this constant load for the engine and generator with a varying actual load of course means there must be some secondary or "dummy" load. The dummy load would be non-productive as distinguished from the actual, real, or productive load. With this as a start, the constant load would be obtained in the following manner:

If the engine were capable of one-half horsepower, the dummy load would have a range of zero to one-half horsepower. Under a condition of zero real or productive load the dummy load would contribute all or one-half horsepower to the total load. For a condition of one-half horsepower real load the dummy load would contribute nothing to the total load on the engine. Any real load between the limits mentioned would be

supplemented by an appropriate amount from the dummy load to provide a load of one-half horsepower for the engine. The combination of real and dummy loading could be designed for any total load and consequently could be used for any size engine and any per cent of load desirable for any engine.

This method of control offers the advantage that the engine and generator need only be designed to operate at one load. For the generator this is a noteworthy advantage, however, for the engine this still means the engine must be capable of delivering its rated horsepower any and all the time it is in operation. For a stationary application at one elevation and not too great a temperature variation this should be as easy a design problem as could be expected for an engine. If the operation, however, involves varying altitudes and extreme temperatures then additional controls would be necessary for the engine unless some suitable load adjustment provision could be designed into the dummy load.

5.5 CONCLUSIONS

In reviewing and assimilating the control methods, the object of control of the engine-generator package should be kept in mind. That is any comparisons, advantages, or disadvantages should be viewed in the light of how the over-all package performance, size, weight, etc. will be influenced and not merely how the engine or generator might be effected. Further, no attempt has been made in the control phase to design a complete system for sensing deviations and actuating mechanism for corrective action. Therefore, any comparisons presented would still need to be further modified for any additional equipment required for the complete system.

The question of weight saving is difficult to answer without designing the complete system; however, there are a few general comparisons that can be made.

Since the total fuel consumption variation was relatively small for all of the engine variables tested, there would be virtually no weight saving from operating the engine at part load as compared to operating at rated load. This would mean, thus, that the use of a "dummy" load control system would not lead to a heavier package for a specified time of operation, i.e., the fuel required for rated load operation was nearly the same as that for part load operation and so the fuel required for a given time of operation would be similar and so the weight of this fuel required would be similar.

The final weight analysis therefore necessitates the design of all the electrical, electronic, and mechanical devices to accomplish the job before the actual comparison. Since the design of this entire system was beyond the scope of this project, a complete weight analysis is not possible.

The sensitivity that can be achieved would, of course, be dependent on the sensing and relaying components as well as the control variable. However, the "dummy load" method could be adjusted to provide the response

time desired. If the engine control variables were used to maintain generator speed, then the response time would depend on the variable. In general, the variable giving the greatest power change per unit of variable change would provide the greatest sensitivity.

In general, the engine-generator package should be controllable by conventional means. The matter of using a "dummy load" should be considered unless the engines can be developed to a point where their fuel consumption shows marked variation with load.

6. NOISE REDUCTION

Only work of an exploratory nature was conducted in the area of noise reduction since it was felt that the greater efforts should be utilized on the studies of performance and reliability. A literature study (reference 1,13,16,27) and tests were used, however, to provide the basis for the information presented here.

6.1 GENERAL

Although noise is classified as an alternating pressure wave traveling through a medium, it might more simply be referred to merely as "unwanted sound". The components of noise are intensity, frequency, and quality.

The unit of sound level is the decibel which is a relative measure of sound above the hearing threshold. Although the exact limits for noise level to cause injury is still vague, it is agreed by some authorities that damage to hearing is likely to occur at a noise level about 90 decibels.

As a guide to the general intensity level, the following breakdown is presented (reference 13, 16):

110 Decibel	thunder and artillery firing
100 Decibel	large steam whistle, subway
90 Decibel	newspaper press room, elevated train
60 Decibel	average factory, home radio
30 Decibel	public library, quiet office
10 Decibel	quiet church
0 Decibel	sound proof room

6.2 ENGINE NOISE

Since an internal combustion engine is a complex machine, it has unlimited possibilities for creating noise. This noise may be created directly in the air streams passing through the intake or exhaust system or the noise may be created by the vibrating mechanical parts of the structure.

Mr. W. P. Green (ref. 13) reports that in almost all internal combustion engines of a type used in power units the exhaust noise component constitutes the largest component of the total engine noise and the intake noise components the next largest. The miniature engines investigated on this project, however, appear to give a noise component from the metallic parts that approaches those just mentioned.

Induction and exhaust noise differ in that the exhaust noise is due to an explosion from the cylinder. Comparatively then, the pressures obtained are much higher in the exhaust and therefore the high pitched sounds come from the exhaust with the low pitched sounds originating in the intake.

6.3 MUFFLER TESTS

To obtain an indication of the degree of silencing that might be accomplished easily, a commercial muffler and a simply designed test muffler were subjected to several exploratory tests. These tests included pressure drop through the muffler and the influence of the muffler on engine performance.

The commercial muffler incorporated a fiber cylinder about 3/8-inch thick enclosed in a perforated metal cylinder. Various sizes are obtained by utilizing various diameters and length. To one end of the cylinder is attached a pipe for connection to the exhaust system of the engine.

The test muffler, Figure 50, consisted of little more than an expansion and contraction in the exhaust system. In fact, initial tests were conducted with the inner baffle and short tube removed and these components were then added to attempt an improvement in noise reduction. The two tubes, A and B, were each drilled with sixteen 1/8-inch holes to permit the gases to flow from the left chamber to the right chamber.

The commercial muffler was first subjected to flow rates simulating its use in the induction system and exhaust system to determine the pressure drop. Following these tests the muffler was drilled to lessen flow resistance at its minimum cross section and a second set of tests were conducted. The results of these tests are shown in Figure 51. It should be noted that the drilled hole did lessen the resistance to flow and that when the muffler was used in the exhaust system (outward flow) its pressure drop was less than when this muffler was used in the intake system.

Similar pressure drop tests conducted on the test muffler gave pressure drops that were small enough to be neglectable at the 2 cfm flow rate normally associated with the engine.

The Mark II engine, operating at 12,000 rpm was then tested using the three muffler arrangements previously described. With no muffler on the engine, the sound level at 2 feet from the engine was 105 decibels. The commercial muffler reduced this reading 4 decibels when used on the exhaust system. The test muffler, utilizing only expansion and contraction in the exhaust system, reduced the sound level 3 decibels and when similarly used with the baffle and two small tubes in place, reduced the sound level 5 decibels.

No attempt was made to use the test muffler on the intake system, but attempts to use the commercial muffler on this system resulted in erratic engine performance. It might further be pointed out that, although the

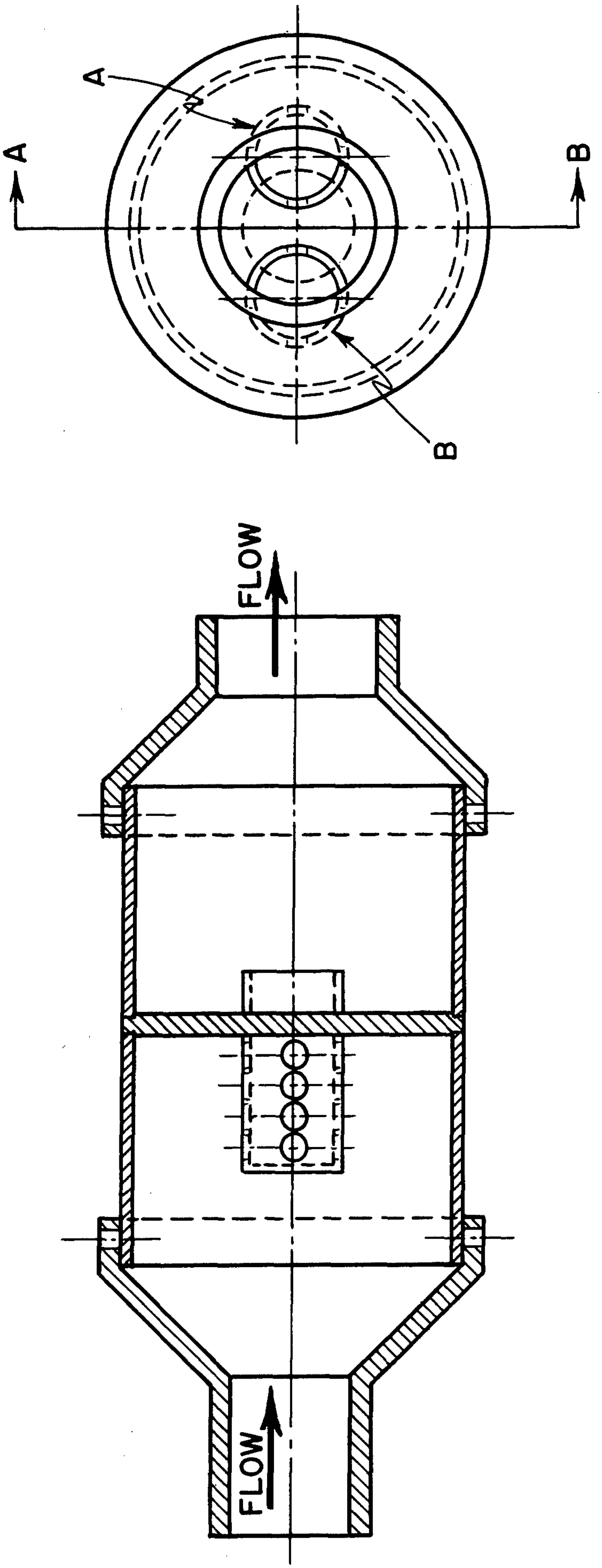


Figure 50. Test Muffler

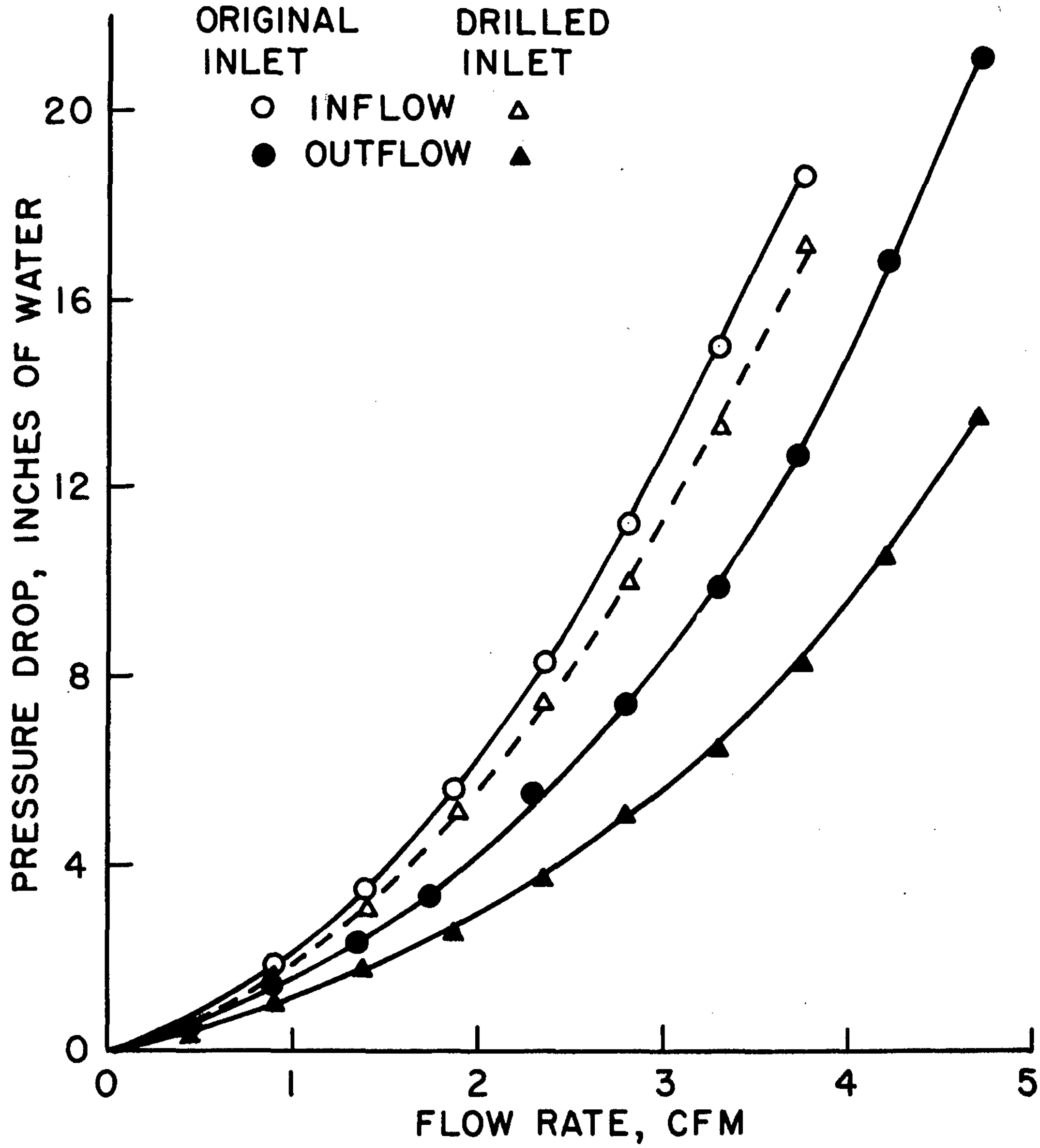


Figure 51. Pressure Drop Through Commercial Muffler

sound level was only reduced a few decibels, many of the objectionable sounds, namely, the high pitched ones, were reduced to a point where they were not nearly so objectionable to the ear.

6.4 CONCLUSIONS

It would seem doubtful if fiber mufflers would be satisfactory for exhaust noise reduction, not only because of the relatively ineffective noise reduction achieved in the exploratory tests, but also because of the early fouling of the fiberous material which would result from the fuel and oil entrained in the exhaust gases.

It is felt, however, (ref. 13) that if size and weight are no object, the noise levels can be reduced to any value desired. This means the definite noise control specifications would be needed by the designer and that imposing any rather severe demands on noise level could result in a heavy, bulky package.

7. RECENT ENGINE EXPERIENCE

To complete the analysis of the effect of compression ratio on miniature engines, the Ruckstell-Hayward loop-scavanged engine was subjected to variable compression ratio and this information is gathered under the heading in section 7.1.

Experience has also been gained in operating the two engines designed and fabricated since the last report. These engines are the Mark III and Mark IV and this experience plus performance curves and engine dimensions are reported in section 7.2.

7.1 VARIABLE COMPRESSION TESTS

The results of varying the compression ratio on the Ruckstell-Hayward loop-scavange engine are presented in Figures 52 and 53. These curves indicate that the optimum compression ratio is in the neighborhood of 9 to 1.

These tests confirm the results obtained earlier on other engines (ref. 35) that, unlike larger engines, miniature engines have an optimum compression ratio. As discussed in the earlier report from this project, this optimum compression ratio is apparently related to a quenching effect of the flame in the very small clearance volumes. It might further be reiterated that a compression ratio of this magnitude is too low to provide good compression ignition; and the small clearances necessary to obtain high compression ratios for good compression ignition quench the flame and permit the gases to cool as soon as they autoignite.

7.2 MARK III AND MARK IV EXPERIENCE

The Mark III and Mark IV engines were designed and fabricated to provide additional test vehicles for investigating performance. The Mark III engine is a two-cycle, cross scavanged engine, while the Mark IV is a four-cycle, overhead valve engine. Some of the important features of these engines are listed in Table D.

The curves of Figures 54 and 55 show the general performance characteristics of these engines. These curves are similar to those reported earlier for the other engines tested. It is felt, therefore, that the results reported in other sections of this report where these engines were used as test vehicles should be fairly representative of those obtainable with miniature engines.

Difficulty was experienced in the valve timing of the Mark III engine and so a series of tests were conducted, using sleeves in the cylinder, to determine the earliest setting of the port openings that would permit satisfactory operation. Table E lists the series of tests involving changes of port timing.

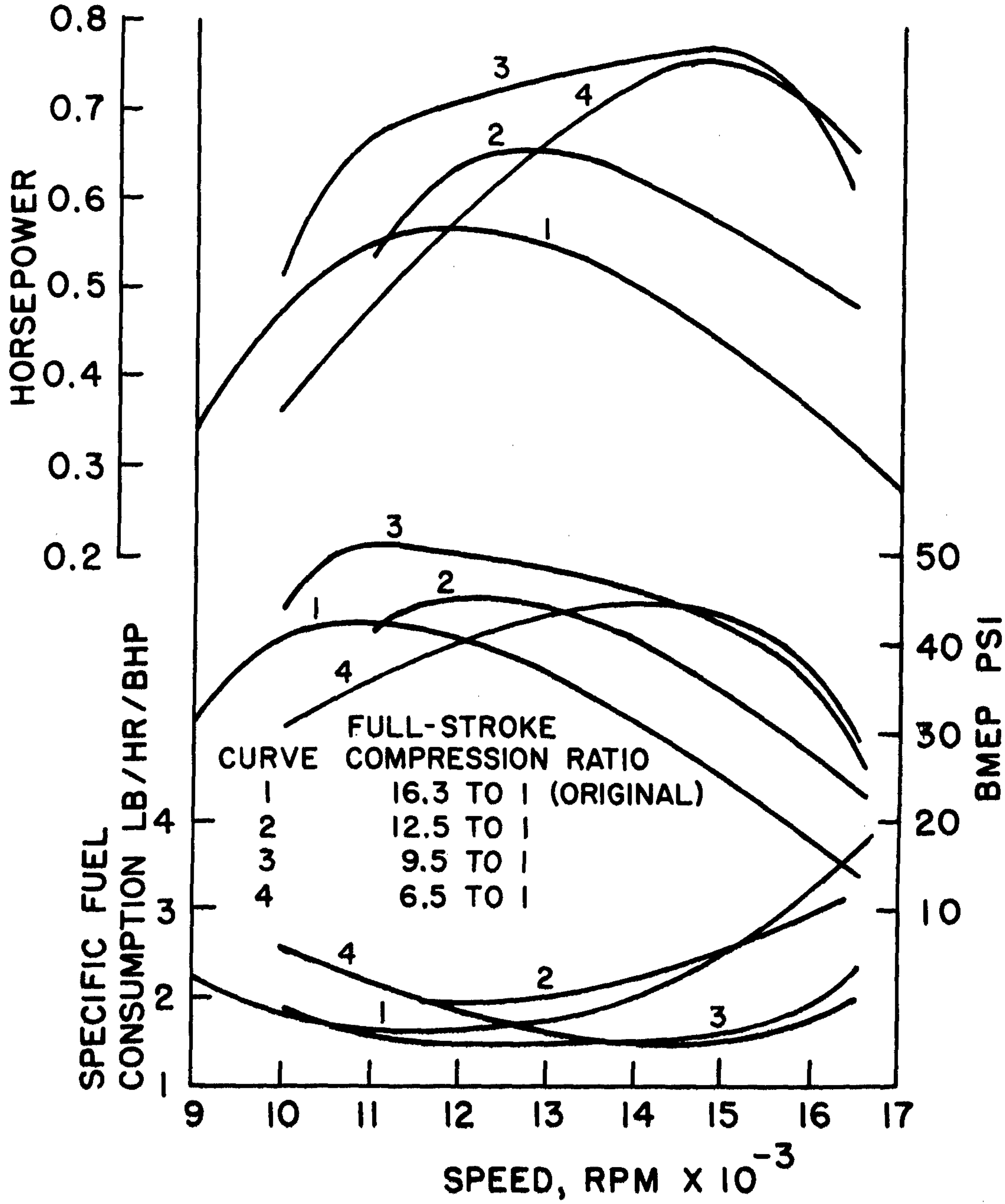


Figure 52. Performance Curves for Ruckstell-Hayward, Loop-Scavanged Engine, Showing Effect of Compression Ratio

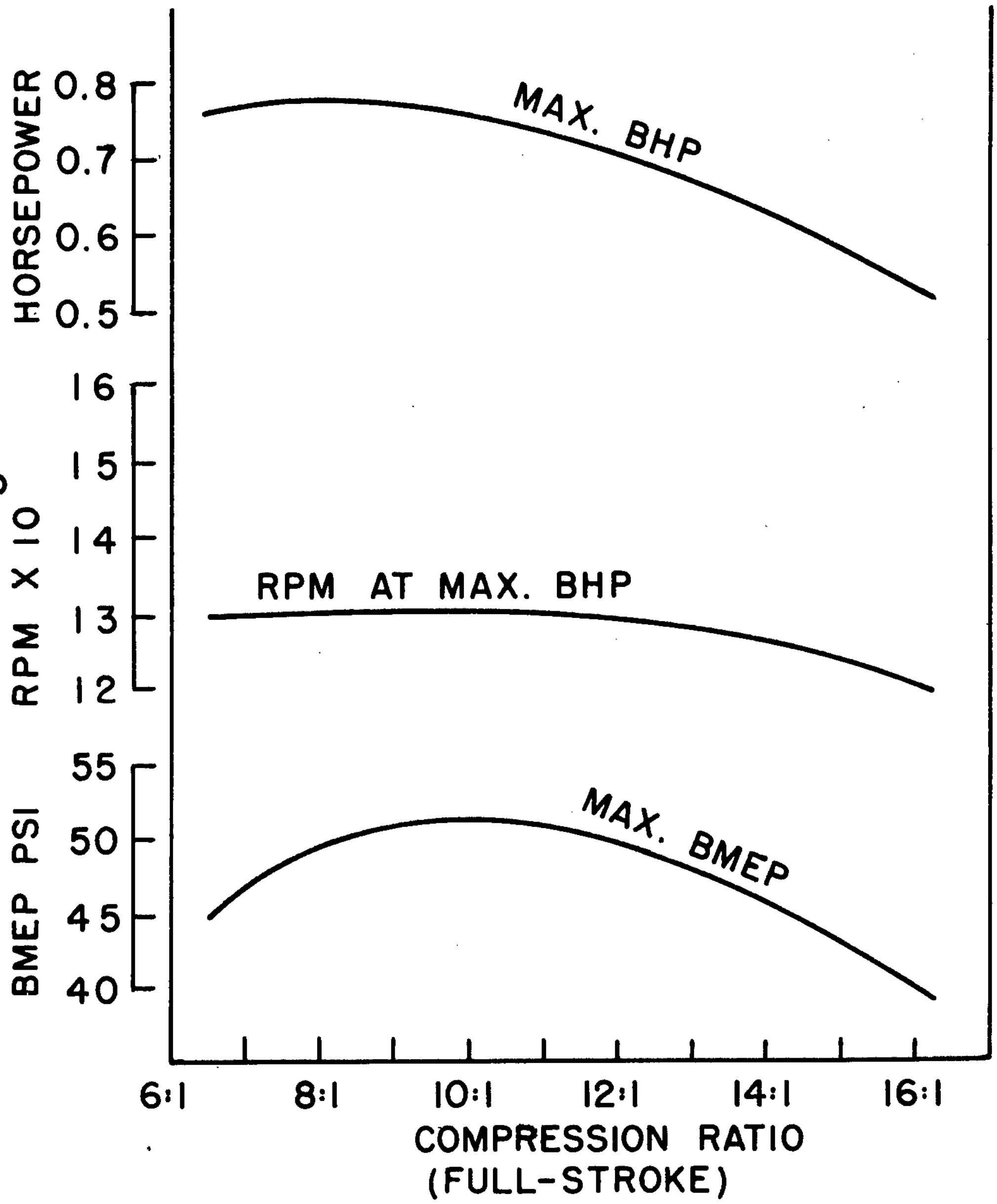


Figure 53. Power, Speed, and Brake Specific Fuel Consumption vs. Compression Ratio for Ruckstell-Hayward, Loop-Scavanged Engine

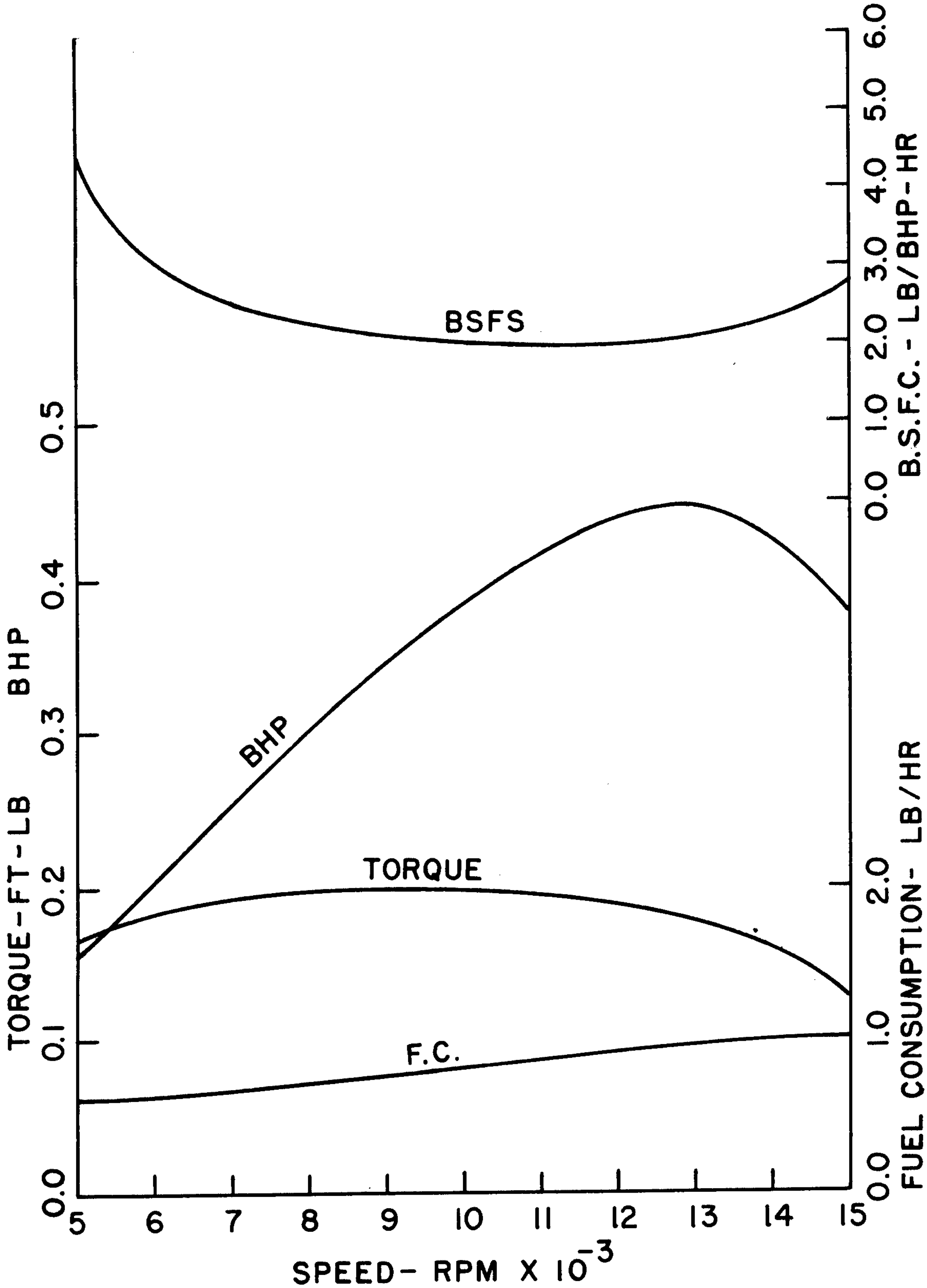


Figure 54. Performance of Mark III

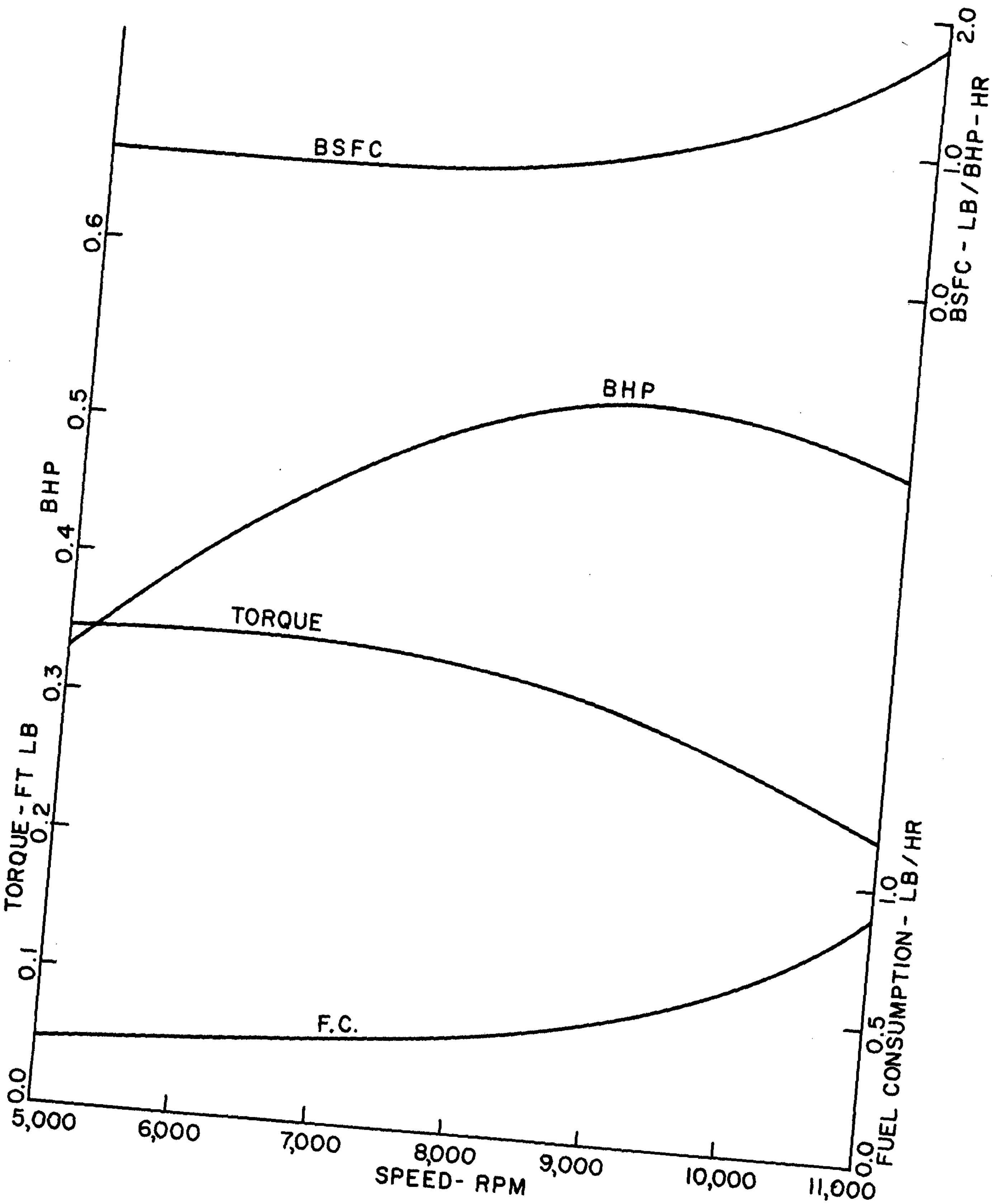


Figure 55. Performance of Mark IV

TABLE D

		Mark III	Mark IV
Cycle		2	4
Type		Cross-Scav.	Ovhd.-valve
Bore, in.		0.890	1.015
Stroke, in.		0.660	0.827
Displacement, cu. in.		0.411	0.669
Bore/stroke		1.350	1.230
Rod length/Crank radius (l/r)		3.790	3.630
Rod bearing type		Roller	Roller
Main bearing type		Ball	Ball
Crankpin, dia/bore		0.292	0.255
Crankpin, proj. area/bore area		0.329	0.302
Main brg., dia/bore		0.562	0.616
Exh. port-area/bore-area		0.281	0.095
Inl. port-area/bore area		0.219	0.095
Piston, length/bore		1.125	0.985
Ext. timing, deg.	open	79.5° BBC	42.0° BBC
	close	79.5° ABC	31.0° ATC
Int. timing, deg.	open	63.6° BBC	6.0° BTC
	close	63.6° ABC	76.0° ABC
Crankcase timing, deg.	open	15.0° ABC	N.A.
	close	20.0° ATC	N.A.
Fuel		80-Octane Aviation Gasoline	
Lubricant		SAE No. 70	SAE No. 20
Compression ratio		8.35:1*	10.1:1
Ignition timing, deg. BTC		55°BTC	55°BTC
Carburetion		Needle-valve, 7/16 in. dia. venturi	

* - based on time of port closing.

TABLE I

Test Number	Intake (Degrees Before Bottom Dead Center)	Exhaust (Degrees Before Bottom Dead Center)
1	36	61.5
2	36	68.5
3	46	68.5
4	51	73.0
5	51	80.5
6	61	80.5
7	61	84.5
8	66	84.5

These values reported as "Before Bottom Dead Center" represent the time the ports were uncovered and, of course, correspond to the time the ports were closed "After Bottom Dead Center". Under the conditions of tests 1 and 2 the engine gave no reaction whatsoever. For tests 3, 4, and 5 the engine fired but would not run. At the timing imposed for tests 6 and 7 the engine fired, but was incapable of carrying its own friction load. At test 8, using either an 80-20 mixture of alcohol and castor oil or an 80-20 mixture of aviation gasoline and SAE 20 oil, the engine was capable of developing 0.33 bhp at 10,500 rpm on the gasoline and 0.39 bhp at 16,000 rpm using the alcohol.

Failures of the Mark III were mainly of the crankshaft due to the attempt to maintain a light-weight engine. Further failures were encountered with the shaft since the shaft was designed to incorporate a coupling that would permit flexibility in the test program. On the design of an engine for a given service, this problem could readily be avoided by using a continuous shaft for the engine and generator, thus eliminating the coupling.

The problems encountered in the Mark IV engine again point out the fact that if a miniature engine is to be successful it must be designed and constructed as simply as possible. Trouble was experienced with the chain drive for the cam shaft actuating the overhead valves. Wear was extremely rapid on the sprockets thus causing the chain to loosen and giving improper engine performance. Crankshaft failures were also present on this engine, thus again pointing out that care must be exercised in scaling down the size of the parts. Also, as in earlier engines built under this project, trouble was experienced with shaft failures because the shafts were designed to give added flexibility.

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